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PIPELINE DIVISION

Proceedings of the American Society of Civil Engineers

CONTENTS

March, 1957

Papers

DIVISION ACTIVITIES

Journal of the
PIPELINE DIVISION
Proceedings of the American Society of Civil Engineers

DESIGN AND STRENGTH OF WELDED PIPE LINE BRANCH CONNECTIONS[a]

E. C. Rodabaugh[1] and H. H. George[2]
(Proc. Paper 1193)

INTRODUCTION

When an opening is cut into a pipe the uniform distribution of stress under pressure is interrupted; the resulting stresses around the opening are significantly higher than those in unperforated pipe. Accordingly, when this opening is made for a branch connection, reinforcement is required to obtain strength equal to that of the unperforated pipe and to provide for external loadings which may be imposed through the branch pipe as shown in Figure 1.

As yet there is no theoretical analysis of the strength of pipe line branch connections, the problem being extremely complex due not only to the shape of the structure but also due to the presence of four independent variables—the diameters and thicknesses of the run and branch pipes. Consideration of laterals adds another variable, the lateral angle. Accordingly, the design of branch connections is based entirely on empirical rules derived from experience and tests. The empirical rules generally used for the reinforcement of fabricated branch connections in pipe lines are given in the American Standard Code for Pressure Piping, ASA B31.1. Similar rules for reinforcement of branch openings are also given in the ASME Boiler and Pressure Vessel Code.

Brief discussions of loadings, Code requirements and field failures of branch connections are presented. A rather extensive amount of test data has been accumulated and is reviewed and correlated with Code requirements and field experience in the following order.

1. Pressure loading
   a) Static pressure
   b) Cyclic pressure

Note: Discussion open until August 1, 1957. Paper 1193 is part of the copyrighted Journal of the Pipelines Division of the American Society of Civil Engineers, Vol. 83, No. PL 1, March, 1957.

a. Presented at a meeting of the Committee on Pipelines, Construction Division, ASCE, October 16, 1956, Pittsburgh, Pa.

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2. External loadings
   a) Static external loads
   b) Cyclic external loads

Finally, the results of the test data and field experience are summarized in a general way under "Recommended Design Practice."

Loadings on Branch Connections

The types of loadings that may be applied to a branch connection are illustrated in Figure 1. Generally in field service a combination of these loads will be applied.

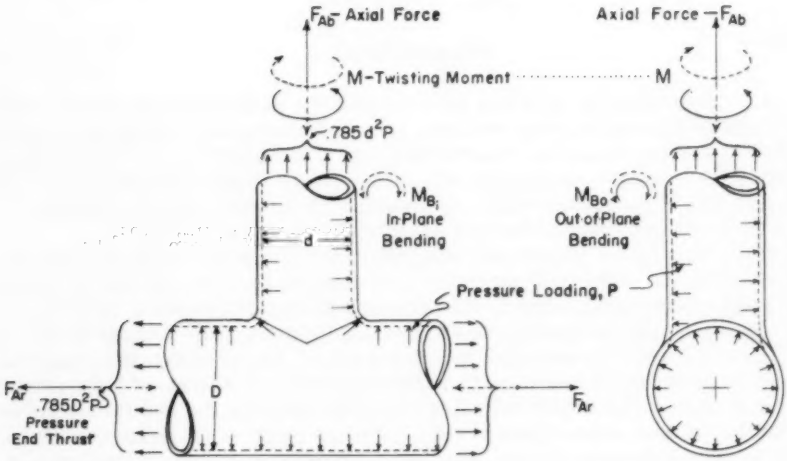


Fig. 1. Loadings Applicable to a Pipe Line Branch Connection.

The internal load, i.e. the pressure, exerts a uniform load normal to the pipe walls; this load may be negative in the case of vacuum service. The end thrusts due to the pressure may or may not be present, depending upon how the pipes are anchored adjacent to the intersection. The internal pressure may vary due to such factors as:

a) change in flow rate;
b) intermittent service;
c) pulsations set up by the pump and/or line resonance.

The external loads have been classified as bending, twisting and axial; bending being further subdivided into "in-plane" and "out-of-plane." In service, the bending load may be between these planes; however, for analytical purposes, the actual bending load may be separated into its in-plane and out-of-plane components. Static external loads are imposed by unsupported weight of the piping, installation misalignment, foundation or support settlement and similar factors. Cyclic external loads may be present due to

conditions such as:

a) cyclic temperature changes producing variations in the pipe line length and corresponding variations in loads on the branch connection;

b) mechanical vibrations of the branch line.

Code Requirements

Rules for reinforcement of fabricated branch connections, primarily from the standpoint of providing adequate internal pressure strength, are given by four sources:

1. ASA Code for Pressure Piping, Section 6, "Fabrication Details"
2. ASA Code for Pressure Piping, Section 8, Paragraph 831.4, "Reinforcement of Welded Branch Connections"
3. ASME Unfired Pressure Vessel Code, Paragraphs UG-36 to UG-46, "Openings and Reinforcements"
4. ASME Power Boiler Code, Paragraph P-268, "Openings and Reinforcements"

The basic concept of reinforcement of branch connections is illustrated by Figure 2. Essentially, these rules require that the metal in the run-pipe cut out by the opening must be fully replaced by extra metal placed within a zone close to the opening. Details of reinforcement requirements and restrictions on the opening size, ratio of opening to header size, angle between header and branch, etc. vary with the specific Code. All of the Codes point out that the rules provide reinforcement for internal pressure; external loadings are not evaluated or are assumed to be those normally applied to such connections. Cautionary notes are given requiring special consideration where the external loads are unusually high or may be cyclic. Special care is also required where the pressure may be cyclic.

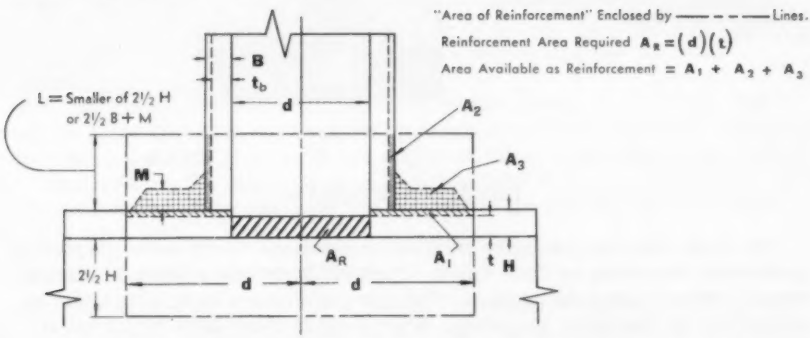


Fig. 2. Typical Code Reinforcing Requirements.

Branch connections which are sold as "Steel Butt-Welding Fittings" are covered by American Standard ASA B16.9. In this standard, adequate strength of the fittings is insured by means of proof tests; the standard requires that actual bursting pressure of the fittings must be at least equal to the computed bursting pressure of the seamless pipe of the size, schedule number (wall

thickness) and material designated by the marking on the fitting. Reputable manufacturers of such steel butt-welding fittings run numerous hydrostatic tests on their pipe fittings to insure that their design is such as to meet this requirement.

Field Failures and Their Causes

Several failures of saddle-reinforced branch connections on gas and oil transmission lines have been reported during the past few years; a failure of this type is shown in Figure 3a.

Another type of commonly encountered field failure involves a small branch connection to a relatively large run pipe, reinforced by pads or saddles or, in some cases, unreinforced; mechanical vibration or movement of the branch pipe with respect to the run has caused failures of such connections; a failure of this type is shown in Figure 3b. This type of branch connection formerly gave considerable trouble in steam power plants, however, experienced pipe designers in that field now recognize the inherent high stress conditions in such connections and make the small branch pipes of XXS or Sch 160 wall thickness and, in some instances, maintain a certain minimum diameter for small branch connections.

Fig. 3a

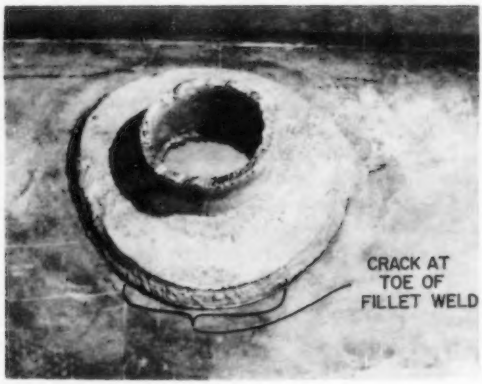


Fig. 3b

Fig. 3. Field Failures.

The field failures generally involved connections which were adequately reinforced according to Code rules. Further, hydrostatic tests on similar branch connections would indicate that the connections were adequately designed for the operating pressure. Since field failures have occurred at normal operating pressures, it appears to the authors that such field failures of branch connections are not usually due to static pressure load but are probably due to such loads as:

a) cyclic internal pressure
b) excessive static external loads
c) cyclic external loads

In considering the possible causes of field failures, other factors besides the loads must be considered. A fabricated branch connection involves a considerable amount of manual welding which introduces the welding quality as a factor. The weld around the intersection between the run and branch pipes is rather difficult to make with consistently good quality, even for an experienced pipe welder. The ends of the branch pipe and/or the edge of the hole in the run pipe must be carefully beveled if a full penetration weld is to be obtained. In welding a pad or saddle type reinforcement, the fillet weld between the edge of the pad or saddle and the run pipe can result in an undercut at the toe of the fillet weld in the run pipe and, as will be shown later, this is a favored location for failure in such reinforced connections. Another factor, in connection with welding, is the possibility of starting cracks by welding. It is known that, in the case of steels rather high in carbon and manganese welded at low ambient temperatures in a constrained position, welding may start cracks either in the weld metal itself or in the adjacent base material. While the authors are not aware of any field failures directly attributable to weld associated cracking, it cannot be entirely ruled out as a possible contributor to field failures.

Other factors which may influence or cause field failures are corrosion and, at high temperatures, creep or graphitization.

Static Pressure Tests

Static pressure tests may be briefly described as tests wherein the pressure is gradually increased until the branch connection fails by yielding and rupture. In conjunction with these static tests, additional information is gained by measuring the strains at critical locations as functions of the pressure. These strains, which may be measured by electrical resistance or mechanical strain gages or by brittle coating, may then be converted to stresses and compared with allowable stresses for the material under static or cyclic loading conditions. Either strains or overall dimensional changes may be plotted against the applied pressure to determine the pressure corresponding to yielding; however, in the case of a complex shape such as a branch connection, the practical significance of yielding has not been evaluated.

It is impossible here to describe and give the results of all of the static pressure tests that have been run on branch connections. A list of references to published test data is given in the Appendix, along with a brief abstract of their contents. In the following we will cover typical tests on the more widely used types of branch connections and indicate in general terms the significance of such tests.

a) Unreinforced Branch Connections

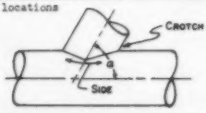
For the present purposes, an unreinforced branch connection will be defined as a branch connection made by welding a branch pipe to the opening in the run pipe without the provision of additional reinforcement. It is possible that such a connection may be fully reinforced according to Code rules. This occurs where the branch pipe is very thick in relation to its diameter; also, if the run pipe is twice as thick as is required to hold the design pressure, the branch connection is generally fully reinforced for the design pressure according to Code rules.

Figures 4a and 4b show failures produced by hydrostatic tests to rupture i.e. burst tests. The left hand photographs show failures in the "crotch" of the intersection, which is the more normal location of the rupture; however, rupture often occurs at the sides as shown by the right hand photographs; the intersection weld being a weak point. Table 1 describes and gives the results of a number of typical burst tests of unreinforced branch connections.

Table 1: Typical Test Results, Burst Tests of Unreinforced Branch Connections

Description					Burst Pressure, psi P_a	Failure Location[1]	Ratio[2] $\frac{P_a}{P_c}$	Tests by:
Run		Branch		Angle, α degrees[1]				
Outside Diam., in.	Wall, in.	Outside Diam., in.	Wall, in.					
4.500	.237	1.315	.133	90°	6350	Run Pipe	1.0	Tube Turns
"	"	2.375	.154		6175	Crotch	.97	
"	"	3.500	.216		6100	Side	.96	
"	"	4.500	.237		5300	Crotch	.83	
4.500 (Range of 8 tests)	.237	4.500	.237		5350 to 6200	7 in crotch, 1 in side	.73 to .85	
Various sizes of straight tees					----	----	.6 to .8	Blair (1)
12.500	.281	10.500	.250		2200	Side	.79	
6.500	.176	4.500	.144		2920	Side	.86	
12.500	.687	6.625	.432		5850	Crotch	.91	Seauloom (2)
24.000	.312	4.500	.237		1970	Crotch	.90	Abrahan and McClure (3)
"	"	12.750	.250		1580	Crotch	.72	
"	"	24.000	.312		1620	----	.74	
1.900	.127	1.900	.127	45°	8050	Crotch	.86	Tube Turns
6.625	.191	6.625	.191		2750	Crotch	.68	
"	.237	"	.237		2570[3]	Side	.50[3]	
"	.500	"	.500		5200	Crotch	.51	
Various sizes of straight laterals, 60°				60°	----	----	.5 to .7	Blair (1)
Various sizes of straight laterals, 45°				45°	----	----	.4 to .6	
12.500	.281	10.500	.250	60°	1850	Side	.66	
"	"	"	"	45°	1230	Side	.44	
6.500	.176	4.500	.144	60°	2240	Side	.66	
"	"	"	"	45°	2000	Side	.59	

[1] Definition of α and failure locations



[2] P_c is the actual burst pressure of an equivalent straight pipe, when available, otherwise it is the calculated burst pressure of equivalent pipe using the Barlow formula with run pipe dimensions and ultimate tensile strength of run pipe material.

[3] Average of three tests.

Based on hydrostatic burst tests of unreinforced tees and laterals, an empirical formula for the strength of such branch connections has been developed which is:

$$\frac{P_a}{P_c} = 1 - \frac{d}{d_r}\left(1 - 0.7E \sin^{1.5}\alpha\right) \qquad (1)$$

where P_a = burst strength of branch connection, psi

P_c = burst strength of uncut pipe of same O.D. and wall thickness as the run of the branch connection, psi

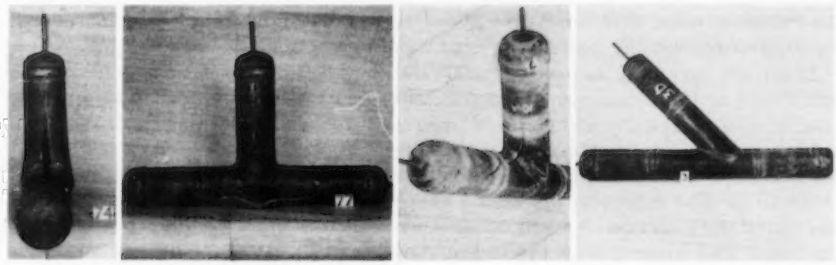
d = branch inside diameter, in.

d_r = run inside diameter, in.

E = weld efficiency; $E = .8$ for field welding; $E = 1.0$ for shop welding where procedures for insuring good quality welds are provided.

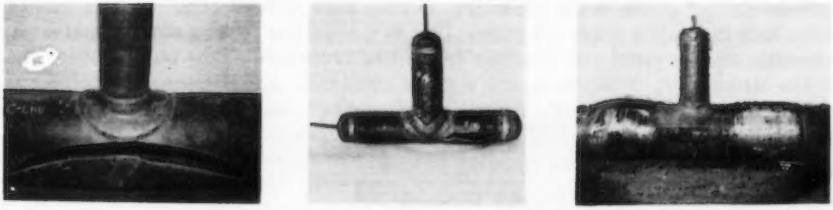
α = angle between run and branch (acute angle for laterals)

In addition to hydrostatic tests to rupture, a number of tests have been made on unreinforced branch connections in which stresses around the branch were determined by means of strain gages. As might be expected, stresses in an unreinforced branch connection are highest at the junction between the branch and run pipe, the stresses rapidly damping out with distance from the



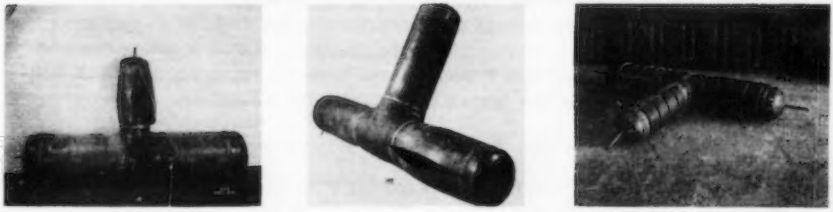
4a; UNREINFORCED TEES 4b; UNREINFORCED 45° LATERALS

FABRICATED UNREINFORCED BRANCH CONNECTIONS



4c; SADDLE REINFORCED 4d; PAD REINFORCED 4e; ENCIRCLEMENT SADDLE REINFORCED

FABRICATED REINFORCED BRANCH CONNECTIONS



4f; MANIFOLD FITTING 4g; FORGED TEE 4h; FORGED TEE (Legs Reinforced)

INTEGRAL REINFORCED BRANCH CONNECTIONS

Fig. 4. Failures in Static Pressure Tests.

intersection. Variation of stresses around the intersection of three unreinforced branch connections are shown in Figure 5, abstracted from Reference 3. The ratios of maximum measured stresses to the nominal calculated stresses are plotted against the position around the intersection.

In the absence of a theoretical analysis, it is difficult to draw any general deductions as to stresses in unreinforced branch connections. It should be noted that stresses determined by strain gage measurements are valid only as long as the structure behaves elastically. In particular the stresses so obtained do not necessarily give any information as to the burst pressure of the connection. For example, the strain gage measurements of the 24" x 24" fabricated unreinforced branch connection indicated a maximum stress which was about five times the stress in the run pipe at locations remote from the branch connection; however, this particular branch connection, when subjected to a burst test, had about 75% of the strength of the run pipe or a "stress ratio," at the burst pressure, of only 1.33 instead of 5. This is due to the fact that when a branch connection is fabricated from a ductile material, the material can yield and readjust to a more favorable shape before breaking. On the other hand, when loads are applied cyclically the stresses indicated by strain gages are significant. As will be discussed later herein, a rough correlation exists between static strain gage measurements with the number of cycles to failure and failure location in cyclic pressure tests.

b) Fabricated Reinforced Branch Connections

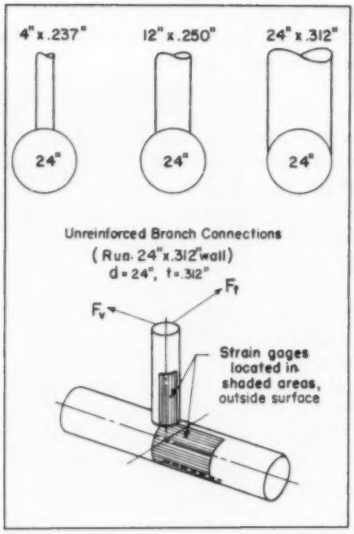
A fabricated reinforced branch connection is defined herein as a branch connection which has additional reinforcement components placed around the branch outlet; these reinforcement components usually consist of saddles, pads, encirclement sleeves or combinations thereof. Table 2 describes and gives the results of a number of typical burst tests of reinforced branch connections. Typical failures are shown in Figures 4c, 4d and 4e.

Table 2: Typical Test Results, Burst Tests of Reinforced Branch Connections

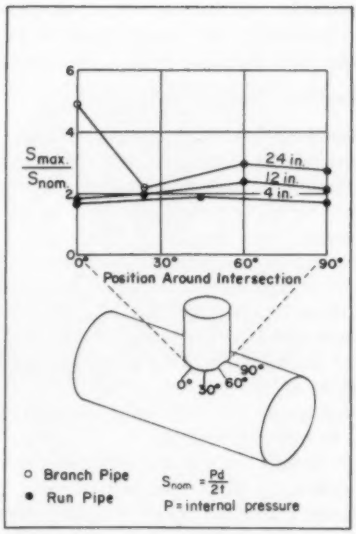
Description[1]				Reinforcement Type[2]	% Reinforcing per ASA Code[3]	Burst Pressure, psi P_a	Failure Location[4]	$\frac{P_a}{P_c}$	Tests by:
Run		Branch							
O.D., in.	wall, in.	O.D., in.	Wall, in.						
4.500	.237	4.500	.237	Pad	80	7150	Run Pipe	.98	Tube Turns
4.500	.134	4.500	.134	Pad	85	3750	Branch Pipe	.91	
4.500	.237	4.500	.237	Saddle	160	6350	Run Pipe	.88	
12.750	.188	6.625	.280	Encirclement Saddle	155	2350	Run Pipe	1.22	
22.000	.312	10.750	.500	Saddle	160	2100	Run Pipe	.99	
6.500	.250	6.500	.250	Pad	85	4800	Run Pipe	.92	Blair (1)
8.625	.500	8.625	.500	Pad	110	8750	Run Pipe	.99	Seabloom (2)
30.000	.344	16.000	.375	Saddle	100	1990	Run Pipe	1.08	Barkow and
"	"	"	"	Pad	105	1865	Run Pipe	1.01	Huseby (4)
"	.500	"	"	Saddle	85	2310	Run Pipe	.93	
"	.500	"	"	Pad	105	2490	Run Pipe	1.00	
"	.344	"	"	Saddle	70	1640	Run Pipe	1.00	
"	.344	"	"	Encirclement Sleeve	120	2060	Crotch	1.12	

[1] All tests were on 90° branch connections (tees).
[2] See figures 7 and 8 for sketches of these types of reinforcements.
[3] Percent of reinforcing required to make intersection equal in strength to the run pipe in accordance with the American Standard Code for Pressure Piping, ASA B31.1-1955, Par. 639.
[4] Failures of pad or saddle reinforced branch connections usually occur on the side at the toe of the fillet weld between reinforcement and run pipe as shown in Fig. 4c.

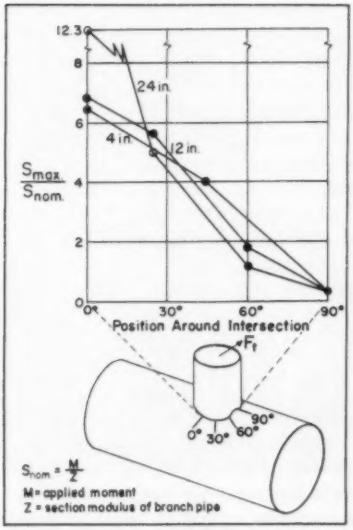
Adequately reinforced branch connections, such as those tested, fail in the pipe at about the pressure at which unperforated pipe would be expected to fail. In other words, tests available to date indicate that a 90° branch connection adequately reinforced in accordance with ASA Code for Pressure Pip-



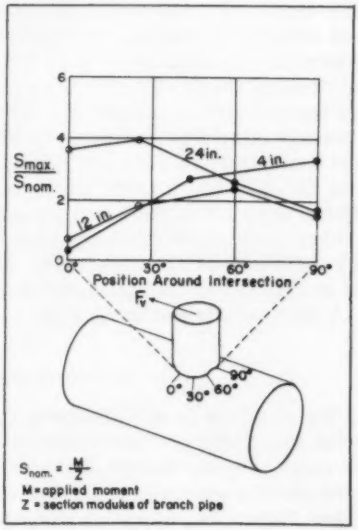
Branch-connection tests.



Ratio of maximum to nominal stress in internal-pressure test



Ratio of maximum to nominal stress in transverse-bend test



Ratio of maximum to nominal stress in axial-bend test.

Fig. 5. Experimentally Determined Stresses in an Unreinforced Branch Connection (Abraham and McClure).

ing rules is equivalent to the run pipe in burst strength. It will be shown later; however, that these types of reinforced branch connections can be much weaker than straight pipe if the pressure is cyclically applied.

c) Manifold Welding Fittings

Manifold welding fittings are used where several branch outlets in a line are needed in close proximity to each other. A photograph of a typical manifold welding fitting is shown in Figure 6. This type of fitting is made by drawing one or more branch outlets in a section of run or header pipe, the header pipe being considerably thicker than required for unperforated pipe; in this design the reinforcement is integral with the fitting. Welding of the branch to the run is accomplished by means of a relatively simple circumferential butt weld thus avoiding the welding difficulties inherent in the complex branch-to-run pipe weld encountered in fabricated branch connections.

The failure resulting from a burst test of a section of a typical manifold fitting is shown in Figure 4f; in this particular case the branch pipe was the weakest member. This manifold fitting, a 10" x 6" size, was designed to be equal to burst strength to 10" x .500" wall, A106 Gr B run pipe. The burst pressure was 5810 psi which is 1.41 times the calculated burst pressure of the equivalent pipe, thereby proving adequacy of the reinforcement in accordance with ASA B16.9 requirements.

d) Butt-Welding Forged Tees

Butt-welding tees manufactured by the authors' company are forged in a manner causing a preferential increase in wall thickness at the locations of highest stress. In addition, the "barrel" shape of the run and the large crotch radii combine to minimize stress concentrations from either pressure or external bending loads. The failure resulting from a burst test of a typical forged tee is shown in Figure 4g. Since the tee is designed to be stronger than the matching pipe, failure occurred in the run pipe in this particular case at a pressure of 2600 psi which is 1.19 times the calculated burst pressure of the 18" O.D. x .375" wall, A106 Grade B matching pipe. In order to obtain information on the burst strength of the forged tee itself, it is usually necessary to either use heavier pipe legs or to reinforce the pipe legs with steel bands as shown in Figure 4h. In this case failure occurred at 5950 psi which is 1.34 times the calculated burst pressure of the 6.625" O.D. x .280" wall, A106 Grade B matching pipe.

Cyclic Pressure Tests[1]

As the result of several failures of saddle reinforced branch connections reported in gas and oil transmission lines, serious thought was directed towards improving the design of the reinforcement of such openings. Since failures usually occurred at the toe of the weld between the saddle and run pipe (see Figure 3a), attention was directed towards a type of reinforcement which would completely encircle the run pipe. Several varieties of these encirclement type reinforcements were proposed; these are shown in Section 8 of the ASA Code for Pressure Piping, ASA B31.1, 1955. Some of these reinforcements were designed for application to "hot tap" connections, wherein a

1. Part of the test work described in this section has been previously published in Reference (7).

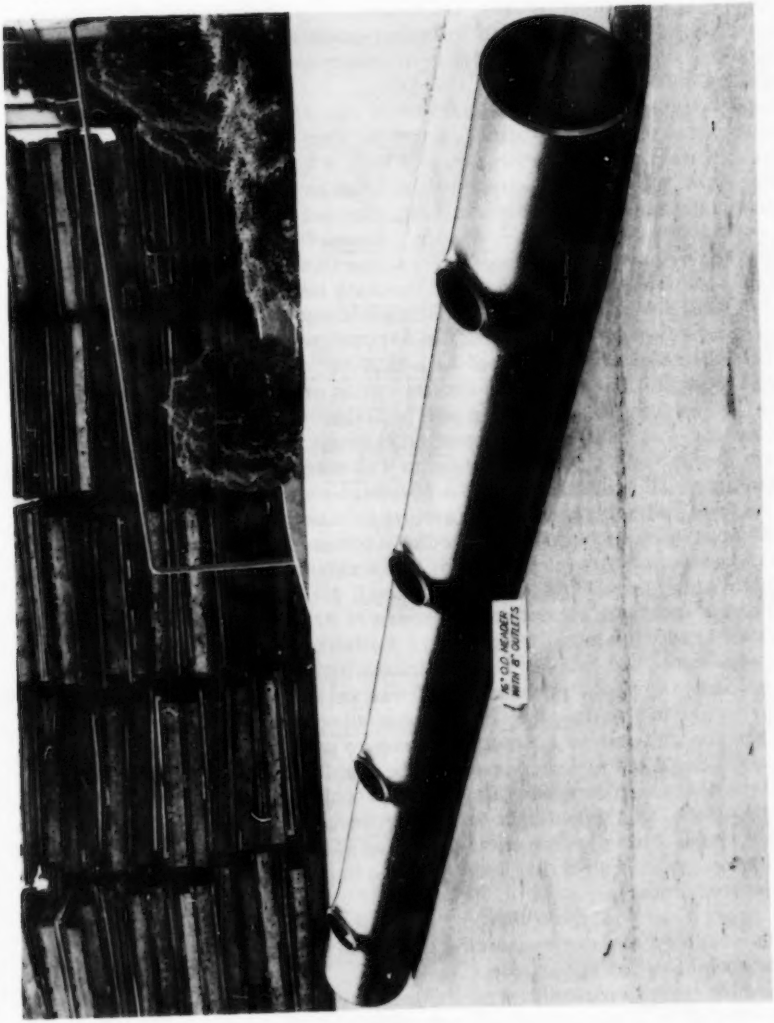


Fig. 6. Typical Manifold Welding Fitting.

branch connection is installed without stopping flow through the main pipe line; they may also be used for normally constructed branch connections. To assist pipeline operators in evaluating the respective merits of these new encirclement type reinforcements, and to generally evaluate the strength of branch connections, the authors' company initiated an extensive test program in which the relative strength of the various branch connections was determined by application of cyclic pressure.

General views and enlarged sections of significant details of the test specimens are given in Figures 7, 8 and 9. Figure 7 shows the first series of test specimens which all involve 22" O.D. x 5/16" wall run pipe and 10.750" O.D. x .500" wall branch pipes. Figure 8 shows the second series of test specimens which involve 24" run pipe with 4", 8" or 12" branches with various reinforcements.[2] Figure 9 shows the third series of test specimens which are sections of manifold welding fittings. In the first series approximately half-scale models of the variants shown were constructed and tested in addition to tests on full scale specimens. The material used in the models was different from the full scale specimens only in that the run pipe was A106 Grade B pipe instead of API-5LX-52 pipe.

The test apparatus developed for the cyclic pressure tests is shown in Figure 10. Briefly, the operation was as follows. The specimen was water filled and air-vented and then connected through a header to the discharge end of a reciprocating pump. The pump was started; as soon as the pressure reached a pre-selected maximum, a pressure control switch opened a by-pass valve allowing discharge into a storage tank connected to the pump suction; as soon as the pressure dropped to a pre-selected minimum, another pressure control switch closed the by-pass valve allowing the pressure to rise again and the cycle was repeated. Each cycle operated a counter. Failure of a specimen caused the pressure to drop below the pre-selected minimum, at which time a low pressure switch automatically shut down the entire apparatus, the counter reading indicating the cycles to failure.

In general, the upper pressure limit was set around 90% of the calculated yield pressure of the run pipe; lines are often tested to this pressure and it might also be attained as a result of pressure pulsations. The lower pressure used was of the order of one-half the yield pressure. While pressure variations of this order of magnitude are probably not frequent in actual service, they were considered necessary to produce failure in a reasonable length of time. This experimental technique of applying intensified loading is, of course, widely used and accepted as a basis of obtaining data on comparative performance.

The tests were continued until failure of the test specimen occurred or until a multiple of the cycles causing failure of a corresponding saddle reinforced connection had been applied.

A first quantitative yardstick of the relative performance of the different variants tested is given by cycles each sustained before failure; these are shown on Figure 12. Since several sizes were involved, it is perhaps more informative to relate the cyclic life of the individual variants to that of a sad-

2. Series II specimens were used at Battelle Memorial Institute for comprehensive strain gage tests prior to being subjected to cyclic pressure tests by the authors' Company. This was part of an extensive investigation of stresses in pipeline branch connections sponsored by the American Gas Association.[8]

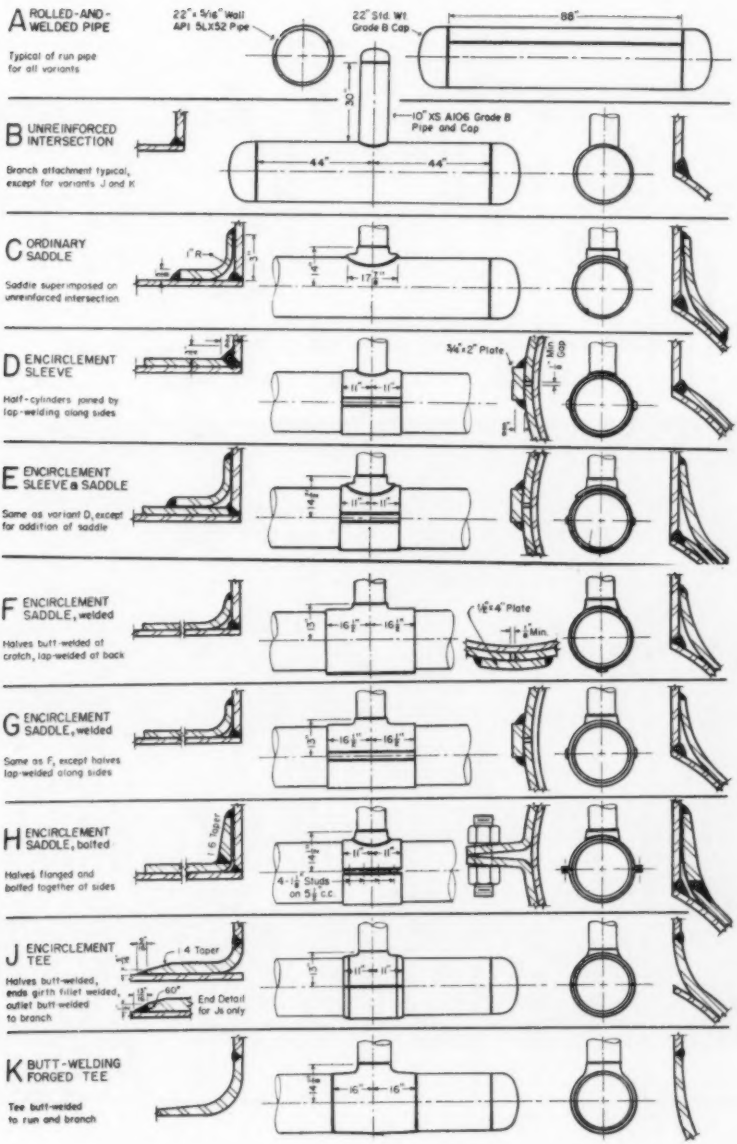


Fig. 7. Specimens for Series I Cyclic Pressure Tests.

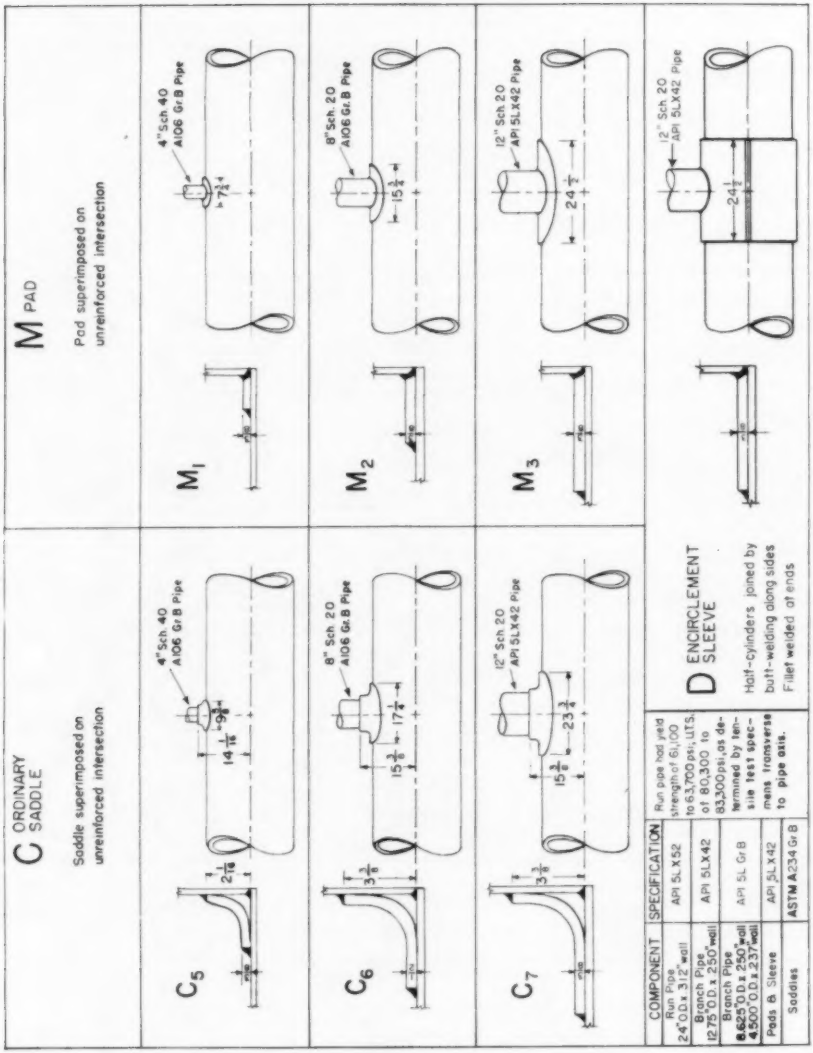


Fig. 8. Specimens for Series II Cyclic Pressure Tests.

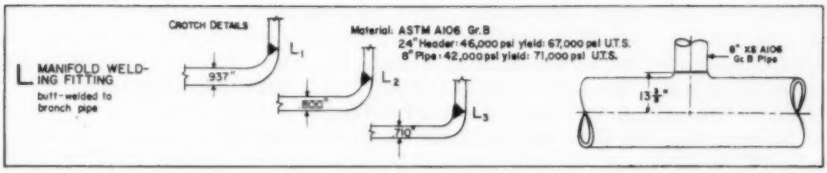


Fig. 9. Specimens for Series III Cyclic Pressure Tests.

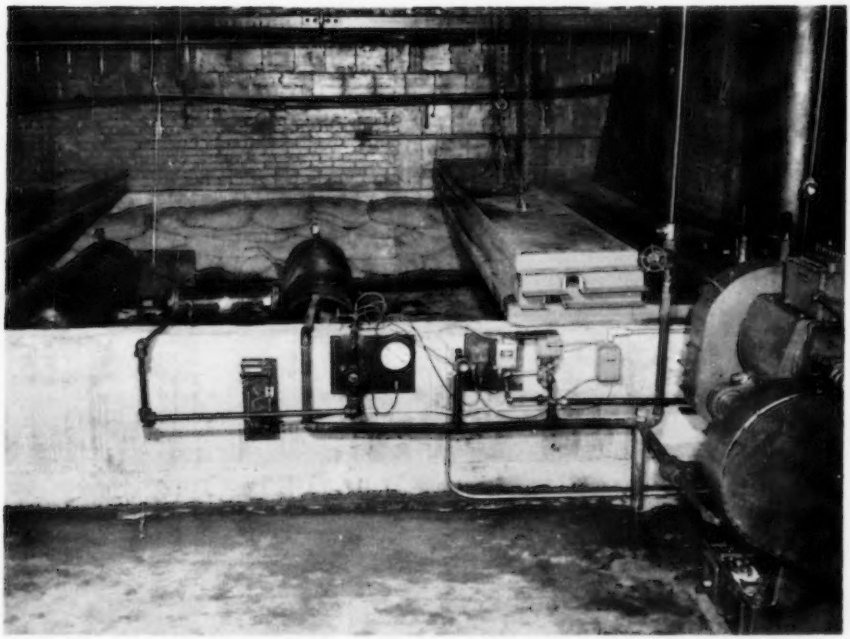


Fig. 10. Cyclic Pressure Test Apparatus.

dle reinforced branch connection. The saddle reinforced connection was chosen as a basis for evaluation since it was the type which had occasionally failed hence this type reinforcement may be considered as a border-line design. Branch connections which are significantly superior to the saddle reinforced design would presumably be adequate under field conditions; those connections which are significantly inferior to the saddle reinforced design might be expected to fail under severe field service conditions.

In order to generalize the available test data on saddle reinforced branch connections, a plot of equivalent stress versus cycles to failure was constructed as shown in Figure 11. The cyclic pressure tests may be considered as applying a mean stress upon which a cyclic stress is superimposed. The relation between completely reversed cycles of stress and a combination of mean stress plus cyclic stress was adopted from the relation suggested by Moore and Kommers.(6)

$$S_e = S_v + 1/3\, S_c \qquad (2)$$

where S_e = equivalent stress for completely reversed cycles
S_v = variable stress component
S_c = constant stress component

Following the type of analysis used for cyclic bending tests,[1] it was found that the relation between stress and cycles-to-failure could be represented by an equation of the type $S_e = CN^{-.2}$; for the particular proportions of saddle reinforced branch connections tested, a value of C = 216,000 fits the test data fairly well.

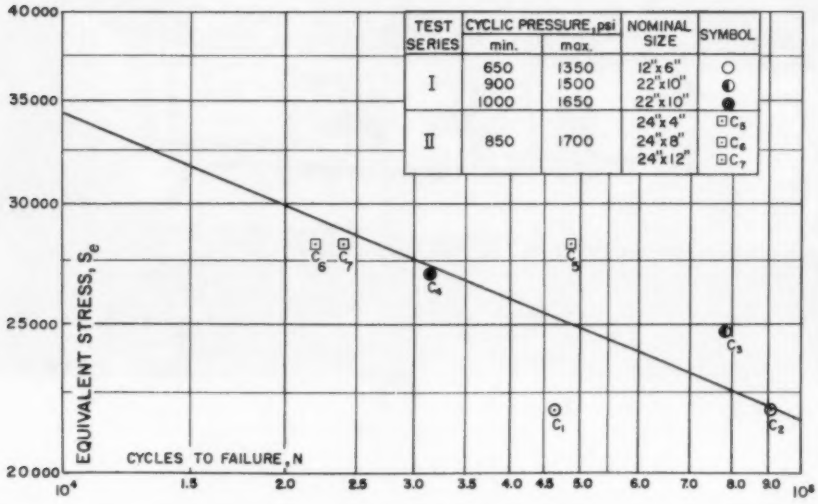


Fig. 11. Saddle Reinforced Branch Connections, Stress vs Cycles to Failure.

1. See page 25.

Of the seven saddle reinforced connections tested; five are roughly geometrically similar. The other two (24" x 4" and 24" x 8", specimens C_5 and C_6), since they involve a smaller ratio of branch to run size, would not, a

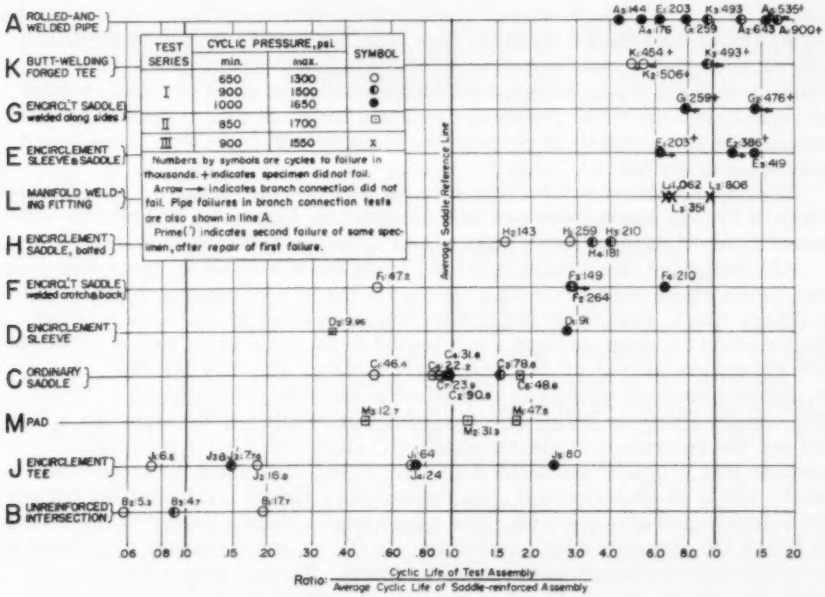
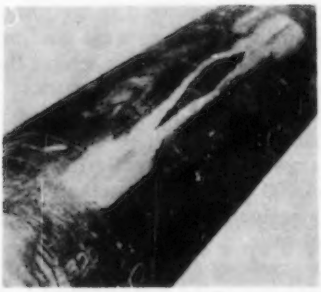


Fig. 12. Comparison of Cyclic Lives of Variants Tested.



13a. General View. 13b. Detail Showing Fracture Along Weld Seam.

Fig. 13. Failure of Rolled-and-Welded Pipe.

priori, be expected to fall on the same S-N curve as the other five specimens. It was found, however, that the one line on Figure 11 fairly well represents the data for all seven specimens. Apparently the controlling factor is the discontinuity at the edge of the saddle-to-run pipe weld, since failure consistently occurred there, and possibly the geometry of the rest of the construction, at least over a limited range, has little influence over the controlling stress. This hypothesis is somewhat supported by the pad reinforced connection tests, which failed at the same location at about the same number of cycles. However, the generalizations are based on rather limited data and extrapolations to shapes or proportions radically different than those tested may not be justified.

Having established a reference line for saddles, the relative life of various types of branch connections are then as shown in Figure 12, wherein the different types of branch connections are arranged in order of decreasing life.

As anticipated, the straight pipe (A) lends itself well as an upper aim point and the unreinforced intersection (B) sets a lower performance limit. The ordinary saddle reinforced connection (C), our yardstick, because it is the construction all other designs are intended to improve upon, lies in between. The forged tee (K) did not fail in a single instance and thus justifies its recognized position as the full equivalent of the pipe it is intended to match. The manifold welding fitting (L), encirclement saddle welded along the sides (G) and the encirclement sleeve-saddle (E) all had cyclic lives essentially equal to that of rolled-and welded straight pipe. The bolted encirclement saddle (H) is an improvement upon the ordinary saddle reinforcement but not the equivalent of straight pipe. The encirclement saddle welded at the crotch and back (F) was somewhat erratic and, on the average, behaves only slightly better than the ordinary saddle reinforcement. The pad reinforced connections (M) and the encirclement sleeve (D) gave about the same performance as the ordinary saddle reinforced connections. Finally, the encirclement tee (J) has been found erratic in performance and generally inferior to the ordinary saddle, contrary to expectations.

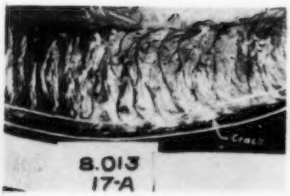
Failures of the rolled and welded steel pipe consisted of cracks in the longitudinal direction in the vicinity of the longitudinal seam weld in the pipe. Generally the split paralleled the seam weld, following one of several shallow grooves observable in the outer surfaces of the pipe. In one failure where the split was short, it was noted that both the length and width of the crack was greater on the outside than on the inside clearly indicating that the failure started on the outside. Figure 13 shows an overall view and close-up of such a typical failure in resistance weld pipe. The large spread in cyclic life between straight pipe specimens, one failing at 144,000 cycles and another running for 900,000 cycles without failure, probably reflects the variation in the severity of the mechanical or metallurgical notches.

The unreinforced connections failed either by cracking across the weld in the crotch, or along it at the side, both types of failure being experienced simultaneously in one assembly; see Figure 14. This is in accord with strain gage test results obtained by other investigators; for connections where the branch size is one-half that of the run or less, the maximum stresses are nearly uniform around the perimeter of the intersection, only the direction changes.

In case of the connection reinforced with an ordinary saddle, the run pipe invariably split along the side at the edge of the saddle as shown in Figure 15. A cross section showing the start of a crack on the side opposite to that where

14a. Arrows Show Location of Cracks

14b. Detail of Crack in Crotch



14c. Detail of Crack in Side

Fig. 14. Failure of Unreinforced Connection.

the rupture occurred is illustrated by Figure 16; this section reveals how a fatigue crack starts at the location of marked chance in section at the toe of the attachment weld, which discontinuity in some instances may be accentuated by under-cutting during welding. Noticeable change in texture of the cracked face evident in the view in Figure 15 indicates how far the fatigue crack progressed inwardly through the wall before this was weakened so that it could no longer sustain the internal pressure and burst open.

The pad reinforced connections failed in the same manner as the saddle reinforced connections as shown by the typical failure in 17.

The encirclement sleeve reinforced connection failed by splitting through the crotch as shown in Figure 18. Both the test variant itself and the failure location bear close resemblance to the unreinforced intersection. By the approximate correlation between stress and cycles to failure suggested by equation (4), a comparison of cycles to failure of the sleeve-reinforced connections with the unreinforced connections implies that the 1/2" thick sleeve in the 22" x 10" assembly served to reduce the controlling stress to 55% of the value without this reinforcement; the 3/8" thick sleeve on the 24" x 12" assembly similarly reduced the stress to 75% of the value without reinforcement. These would appear to be reasonably estimates of the reinforcing effect of the sleeves.

The sleeve-and-saddle reinforcement gave excellent results; two specimens failed in the pipe run outside of the reinforcement and the third in the pipe run under the sleeve, the fatigue crack following the edge of the weld line in the run pipe.

The three variants of the encirclement saddle are very similar to each other in intersection detail. Of these three, only the design wherein the encirclement halves were welded along the sides (G) consistently gave results in line with those obtained for straight pipe. The performance of Variant F, which is distinguished from Variant G by having the encirclement halves joined together by lap weld at the back and a butt weld through the crotches, was somewhat erratic. In the first model test a leak started under the reinforcement at 47,200 cycles; in order to be able to continue the test, both ends of the reinforcement were fillet welded to the run pipe but the assembly only sustained an additional 5,500 cycles before the crotch weld joining the two halves cracked. This somewhat difficult weld had been made in the normal way, without backing. To insure a sound weld, a thin curved backing strip

Fig. 15. Failure of Saddle Reinforced Connection.

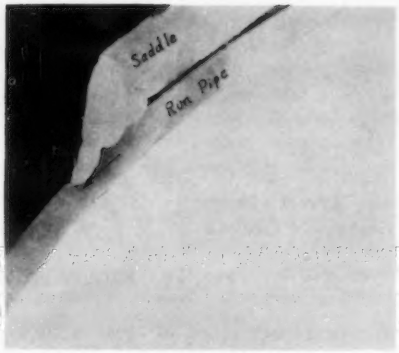


Fig. 16. Saddle Reinforced Connection, Side Opposite Failure Showing Start of Crack.

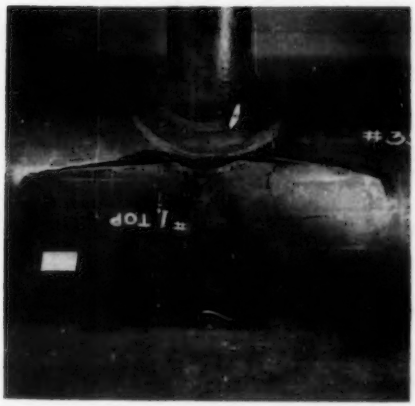


Fig. 17. Failure of Pad Reinforced Connection.

Fig. 18. Failure of Sleeve Reinforced Connection.

was used at this location in constructing the remaining three specimens of this variant. Probably as a result of this provision, the second model ran 264,000 cycles without failure; one of the full scale specimens also showed a fair life, while the other performed only moderately better than the ordinary saddle construction, both failing in the intersection weld

between the run and branch pipe in the crotch area under the reinforcement. A typical failure is shown in Figure 19. Possibly the reason for the difference observed in the performance of variants G and F might be explained by the hypothesis that shrinkage of the two side welds in Variant G in some way produces a better fit-up in the critical zone than is the case where the welds are made at the back and crotch as in Variant F, however there is no direct evidence that this is true.

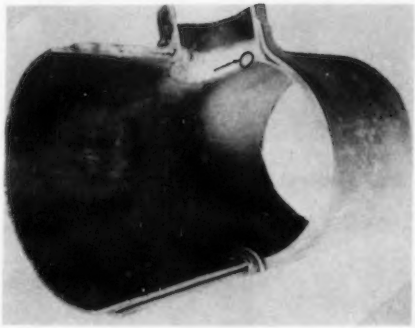


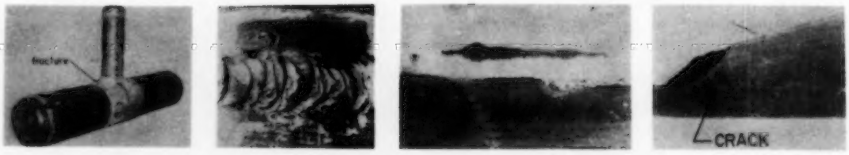
Fig. 19. Failure of Encirclement Saddle Reinforced Connection, Quarter Sectioned to Show Failure on Inside of Crotch.

The bolted variant (H) provided moderate but consistent improvement over the ordinary saddle construction; as in the other types of encirclement reinforcements failure occurred in the intersection weld in the crotch area, see Figure 19. With the thought that close contact of the saddle with the branch-to-run intersection weld might have a beneficial effect, intentional variables in the detailed fit-up in this location were introduced but no marked effect on relative life of the different specimens was produced. This design evolved from the thought that bolts should provide an excellent means of bringing the reinforcement into forceable contact close to the critical intersection weld; in addition, there is no welding against the run pipe such as that which occurs where the sleeve halves are joined by a butt weld. In the design actually tested however, the bolts were placed too far from the shell and the bolting flanges were somewhat too light. Improvements in detail of this design might bring its performance up so as to be comparable with that of straight pipe.

The encirclement tee is basically different than the other reinforcements previously discussed in that the reinforcing member directly carries the internal pressure; the branch pipe is butt welded to the reinforcement and is not welded to the run pipe. This type of branch connection gave unexpectedly poor results. With the exception of the initial failure in the first model, where the butt weld joining the two halves cracked in only 6500 cycles, all failures occurred in the fillet welds between the tee and run pipe, the location in all except one instance being directly in line with the crotch. As can be seen in Figure 20d, the cracks ran in a circumferential direction and apparently started at the root of the fillet weld.

The encirclement tee assemblies, including the fillet welds in all but the last specimen, had been proportioned to parallel the assembly used by Del Buono, Vissat and Williams[9] in their exhaustive and evidently very carefully conducted strain gage tests. It is noteworthy that failure in the cyclic pressure tests occurred in the precise location where these investigators had obtained the highest strains, and that the crack closely resembled that found by them upon sectioning through the crotch plane after bursting their test assembly.

Since the fillet weld quite obviously constituted a point of weakness, the thought naturally presented itself that the performance of this design could be



20a. General View

20b. Close-up Before Grinding Off Weld

20 c. Same as 20b After Grinding Weld, Crack . Brought Out by Penetrant Dye

20d. Cross-Section Through Failure Area

Fig. 20. Failure of Encirclement Tee Connection.

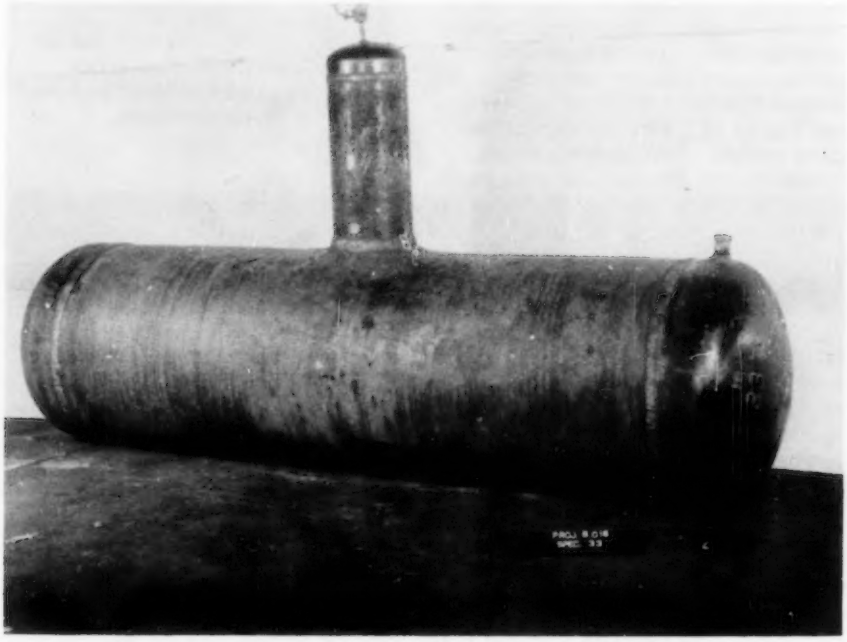


Fig. 21. Failure in Manifold Welding Fitting.

improved by increasing the weld size. Accordingly a special test assembly was made by using what was considered a maximum practical weld size. The weld leg dimension was increased by more than 50% and the tee end was machined with a reverse bevel of 60° and a 3/16" root radius, all of which resulted in over 150% increase in the amount of weld metal deposited. This design with a heavier weld (Specimen J5) lasted 3 1/4 times as long as one with the original weld proportions tested under otherwise identical conditions,

Fig. 22. Combined Static Bending and Pressure.

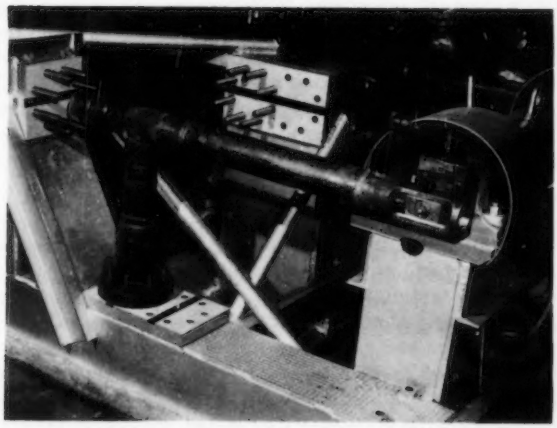


Fig. 23. View of End Bank of Fatigue Test Machine Showing a Fabricated Tee Mounted for In-Plane Bending Test.

however this specimen still had a cyclic life only slightly better than a corresponding saddle reinforced connection.

Finally we come to the forged tee and manifold welding fittings, these being basically different from the preceding variants in that they cannot be used for hot tap connection. In none of the three specimens of the forged tee did failure occur in the tee itself. The two model tests were run to a large multiple of the cycles causing failure in a corresponding ordinary saddle reinforced connection; in the full scale specimen the test was terminated by failure of the run pipe with the tee left unharmed. It is apparent that the integral reinforcement, large crotch radii, preferential thickening in highly stressed areas and gradual change in wall thickness make the forged tee the full equivalent of the pipe it is designed to match.

The manifold welding fittings also gave a good account of themselves but they were not equal to the forged tees. All three manifold welding fittings had a cyclic life of the same order as that obtained for straight pipe. The failures consisted of cracks in the crotch area; a typical failure is shown in Figure 21. It was not expected that the manifold welding fittings would be as good as the forged tees since they do not have the generous crotch radii and preferential thickening in highly stress areas which are obtained with forged tees.

Results of the cyclic pressure tests are correlatable with strain gage tests. In the Series II tests, wherein the specimens had been used for strain gage tests. In the Series II tests, wherein the specimens had been used for strain gage tests prior to the cyclic pressure tests, all seven specimens failed in locations at or near the points of maximum measured stress. As was pointed out earlier, failures of the encirclement tees (J) occurred at the location of maximum measured stress. A rough correlation between stresses and cycles to failure was also observed; since strain gages do not measure the effect of local notches, such as weld defects or undercutting, it would not be expected that good correlation between stress measured by strain gages and cycles to failure would necessarily exist.

Although all of the reinforced fabricated branch connections used in the cyclic pressure tests were more than 100% reinforced by the ASA Code for Pressure Piping Rules, these connections were not all equal to the longitudinally welded straight pipe in cyclic life. Accordingly, while it was concluded that 90° fabricated branch connections reinforced per the ASA rules were equal to straight pipe in burst strength, the same conclusion cannot be drawn for cyclically applied pressure.

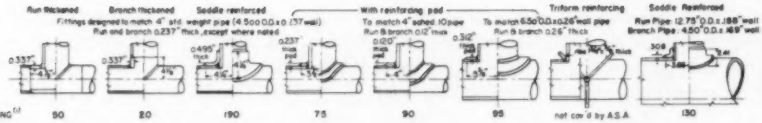
Static External Load

Figure 22 shows a test in which a combination of static internal pressure and static out-of-plane bending load was applied to produce rupture. In this test, a 22" x 10" saddle reinforced branch connection, constructed as shown in Figure 7, was pressurized with water to 1200 psi; then an out-of-plane bending force was applied and gradually increased until failure occurred at a bending moment of 300,000 ft-lb. The pressure corresponded to a circumferential stress in the run pipe of approximately 72% of its yield strength; the bending moment at failure gives a calculated maximum stress in the branch pipe of 73,000 psi. This test indicates that very large static bending loads can be carried by a reinforced branch connection.

Stresses resulting from static bending moments are shown in Figure 5.

Cyclic External Loads

Cyclic bending tests, in which about one hundred different branch connections were subjected to cyclic bending loads to failure, were run using the test machine shown in Figure 23. In these tests, one end of the assembly was anchored rigidly and the other, a hinged end, was forced to deflect cyclically through a given amplitude maintained by the machine eccentric setting. The specimens were subjected to individual load-deflection calibrations, filled with



% REINFORCING[1]	50	20	190	75	90	95	not con'd by A.S.A.	130
STRESS INTENSIF. FACTOR, I								
as computed:[2]	2.84	2.84	1.8	1.97	3.17	2.16	not amenable to this type of analysis	3.70
from tests: in-plane	2.15	2.15	1.10	1.76	2.21	2.58	2.30	1.19
from tests: out-of-plane	—	—	1.08	1.83	2.43	—	2.85	4.18
Ratio: computed / avg. from test	1.32	1.32	1.06	1.09	1.37	0.84	—	1.37

1. Percent of reinforcing required to make intersection equal in internal pressure strength to the run pipe in accordance with the American Standard Code for Pressure Piping, ASA B31.1-1955, Par. 639.
2. See Table 3.

Fig. 24. Reinforced Fabricated Branch Connections Used in Cyclic Bending Tests.

water to provide a ready means of detecting failure, and finally flexed cyclically through a predetermined amplitude until a leak signalled a crack through the entire wall. This was done at several amplitudes for each type of specimen so that upon completion of tests a stress vs cycles-to-failure plot (S-N curve) could be constructed. The stress is the nominal stress in corresponding size and wall thickness straight pipe calculated by the ordinary beam formula:

$$S = \frac{PL}{Z} \tag{3}$$

where S = nominal bending stress, psi
P = applied load (lb), obtained from load-deflection calibration
L = lever arm from point of load application to point of failure, in.
Z = section modulus of corresponding straight pipe, cu-in.

Details of reinforced test specimens are shown in Figure 24; in addition, various designs of forged butt-welding tees and unreinforced straight fabricated branch connections with dimensions as indicated on Fig. 25 were tested. Results of these tests as S-N plots are shown in Figure 25, along with results of bending fatigue tests by Blair.(1) Photographs of typical failures are shown in Figure 26.

Tests conducted by Blair were run with static internal pressure applied to the connections while applying the cyclic bending loads. Accordingly, in these

The data given herein on cyclic bending tests is principally abstracted from reference (5). For more complete details of these tests and their evaluation, this paper should be consulted.

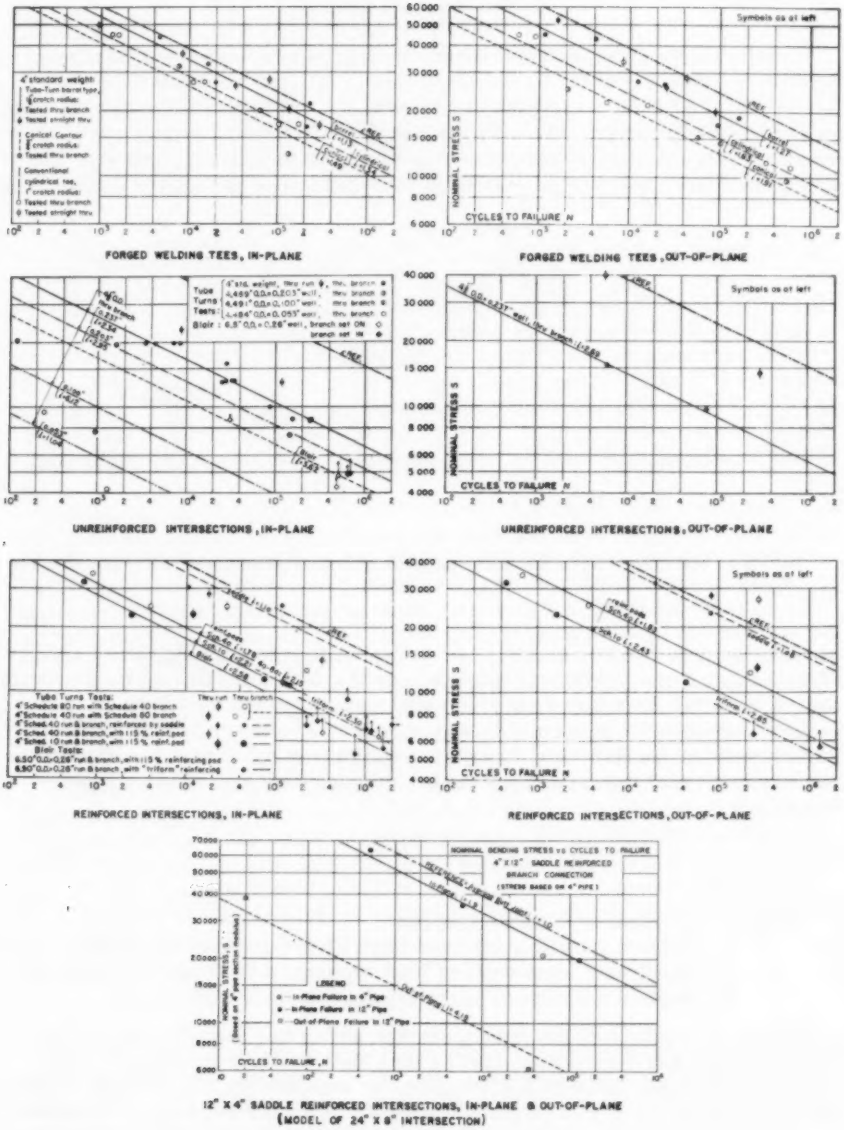


Fig. 25. Cyclic Bending Stress vs Cycles to Failure (S-N) Plots.

tests there was a constant stress due to pressure as well as cyclic stresses due to bending; for these specimens an arrow pointing upward is shown, the tip of the arrow indicating the equivalent stress for complete reversal in accordance with equation (2).

It is pertinent to note the location of failures in the forged tee, indicating the areas of highest stress. It is in just this highly stressed area where the intersection weld of a fabricated branch connection is located. Since it is practically impossible to make such an intersection weld without some defects and notches, such fabricated connections, even though heavier than the seamless forged tee, are liable to premature fatigue failure due to the stress concentrations caused by weld irregularities. Pad reinforced intersections also failed along the intersection weld. In the case of the saddle reinforced connections, the saddle was able to support the intersection weld so that the failure location is transferred to the outer edge of the saddle at the toe of the fillet weld between saddle and pipe. In this location, unless care is used, fillet welding can cause an undercut into the run or branch pipe, thereby creating an additional notch in a highly stressed area. The welding of these tests specimens was done with great care under ideal shop conditions; with carelessly made welds or with poor welding conditions the fatigue life might be very adversely affected.

Similar bending fatigue tests were run on circumferentially butt-welded straight pipe and the results of these approximately 50 tests were also plotted on an S-N graph. The relative strengths of the various types of branch connections were evaluated in relation to the strength of such butt-welded pipe. It was found that the test data could be reasonably well represented by the formula:

$$iS = 245{,}000N^{-.2} \tag{4}$$

where S = nominal applied bending stress in corresponding straight pipe, psi

N = cycles to failure

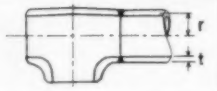
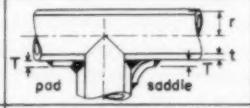
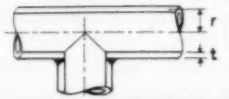
i = stress intensification factor; equal to unity for circumferentially butt-welded straight pipe

The values of the stress intensification factors in relation to butt-welded pipe are shown in Figure 25. In attempting to evaluate and generalize these test results, it was noted that failures in forged tees were analogous in location and direction to those occurring in elbows or curved pipe submitted to the same type of fatigue test. Stresses in curved pipe can be readily evaluated by theoretical methods; accordingly the theory of curved pipe was used as a basis for generalizing the experimental fatigue life data on branch connections, leading to the formulas shown in the table 3.

The stress intensification factors based on these tests have been incorporated in the American Standard Code for Pressure Piping, along with factors derived from similar tests on other piping components such as elbows, mitres and flanged joints.

Each of the individual cyclic bending tests represents considerable expenditure of labor, hence the number of specimens and variants were necessarily limited. Accordingly, the generalizations derived are based on rather limited data; only the pressing need for design information induced the attempt to translate the imperfectly defined pattern of results into working

Table 3: Formulas for Stress Intensification Factors

DESCRIPTION	SKETCH	h	STRESS INTENSIFICATION FACTOR
Full-size Welding Tee per ASA B16.9 (1)		$4.4 \times \frac{t}{r}$	$\frac{0.9}{h^{2/3}}$ but not less than 1
Full-size pad or Saddle Reinforced Fabricated Tee (1) (2)		$\left(1+\frac{T}{2t}\right)^{3/2} \times \frac{t}{r}$	
Full-size Unreinforced Fabricated Tee (1)		$\frac{t}{r}$	

(1) Also use for reducing-outlet tees in absence of directly applicable data.
(2) When T is greater than 1.5 t, use $h = 4.05 \frac{t}{r}$.

constants. As a corollary, extrapolation of the data given to shapes or proportions radically departing from those tested is liable to appreciable error.

Recommended Design Practice

The test data on branch connections accumulated to date represent an immense amount of time and effort; nevertheless, it is not possible to quantitatively apply the test data to branch connections in general. The test data does, however, indicate some general design practices which should be followed.

The design should first satisfy the rules of the Codes cited herein. If service conditions are more severe than contemplated in Code requirements, i.e., if cyclic pressure, high static bending loads or cyclic bending loads of any magnitude may be applied, then consideration should be given to the selection of a suitable type of branch connection which will not only meet Code requirements but will be of optimum design for the expected service conditions.

In an optimum design, one must balance the first cost of the installation against the possibilities of failure and the consequence of such a failure. In some pipe lines which may be shut down with little effect on associated processes, a failure may mean only the necessity of replacing the failed component. In other pipe lines, however, a forced shut down may be extremely costly. Failure of lines carrying lethal or inflammable fluids or gases at high pressure may result in great economic loss and hazard to life.

The test data and field experience all tend to indicate that branch connections with integral reinforcing, large intersection radii, gradual changes in wall thickness and preferential thickening in highly stressed areas will give optimum strength. That this should be true of branch connections is not surprising; parallel experience with machine design has long ago led to the use of generous radii and gradual thickness changes to reduce the possibility of fatigue failures. Branch connections with such characteristics in varying de-

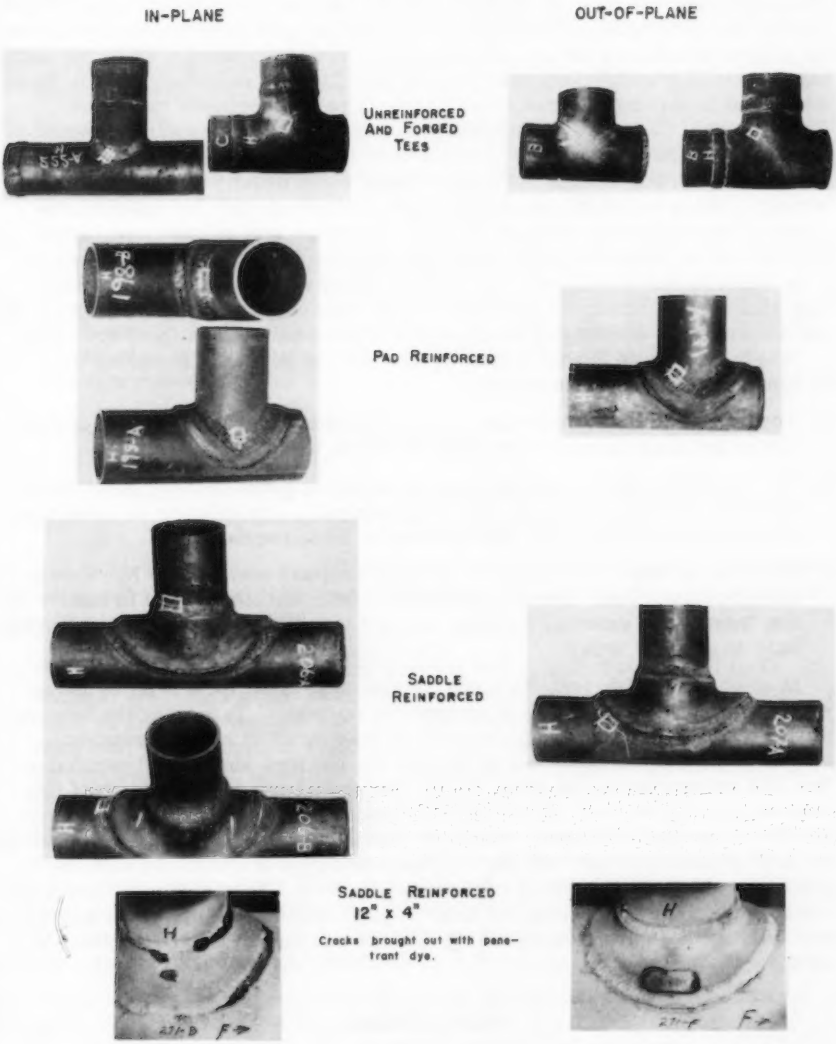


Fig. 26. Failures in Cyclic Bending Tests.

grees are commercially available as forged tee and crosses, manifold welding fittings and drawn outlet laterals. In welding such fittings into a pipe line, care must be taken to obtain a full-penetration weld; however, since the welding is done in a uniform circumferential joint, this is relatively easy to do as compared to making a complicated intersection weld between two pipes. Furthermore, the welds in such integral fittings are not ordinarily located in the area of highest stress.

While integrally reinforced branch connections usually give optimum strength, there are occasions where it is either necessary or desirable to use a fabricated branch connection; e.g. where the size is such that forged fittings are not commercially available, for "hot-tap" connections or where it is believed that the installed cost of a fabricated connection will be less than that using a commercial fitting and that the service conditions, both from the standpoint of severity of loadings and consequences of failure, are mild.

Where fabricated branch connections are to be used, the following types of construction are recommended:

a) For severe service conditions—use a complete encirclement type of reinforcement such as one of the types shown to be adequate by test.

b) For moderate service conditions—use saddles, pads or other reinforcing members designed to meet Code requirements. Sharp angle laterals may require more reinforcing than implied by Code requirements.

c) For mild service conditions—Code requirements may be met by "unreinforced connections," however extreme caution should be used to assure that substantial external loadings will not be applied, even though the pressure is of a low order.

In constructing fabricated branch connections, great care must be taken with welding in order to assure adequate service life. In making the branch-to-run pipe intersection weld, the ends of the branch should be properly shaped and beveled and/or the opening in the run pipe shaped and beveled so that full penetration of the weld can be obtained. Lack of penetration of this weld not only produces a severe mechanical notch in a highly stressed location but also leaves a crevice which is ideal for promoting cell type corrosion, should the fluid have any tendency towards this type of corrosion with the particular metal used in constructing the pipe line. The fillet weld between a reinforcing member, such as a saddle or pad, and the run pipe must also be carefully made to avoid undercutting into the run pipe at the toe of the fillet weld.

REFERENCES

1. "Reinforcement of Branch Pieces" by J. S. Blair, Engineering (London), July 5, Sept. 6, Nov. 29, Dec. 6, 13, 20, 27, 1946.

2. "Bursting Pressure Tests of Welded Pipe Headers" by Eric R. Seabloom, The Valve World, July-August, 1941.

3. "How Stresses are Affected by Branch Connections" by E. D. Abraham and G. M. McClure, Pipe Line Industry, September, 1954.

4. "Welded Tee Connections" by A. G. Barkow and R. A. Huseby; Welding Research Council Bulletin Series, Number 22, May, 1955.

5. "Fatigue Tests of Piping Components" by A. R. C. Markl, Transactions of the American Society of Mechanical Engineers, Vol. 74, No. 3.

6. "Textbook of the Materials of Engineering" by H. F. Moore, McGraw-Hill Book Company, Inc., New York, New York, 1941, p. 57.

7. "Pressure-Pulsation Tests of Branch Connections to Large-Diameter Pipe" by A. R. C. Markl, H. H. George and E. C. Rodabaugh, presented at the American Gas Association Gas Supply, Transmission and Storage Conference, Pittsburgh, Pennsylvania, May 9-10, 1955.

8. "Stresses in Branch Connections," Report of the Supervising Committee for Pipeline Project NG-11, American Gas Association Inc., 1956.

9. "Design of Hot Tap Tee Connections in High-Pressure Pipe Lines" by A. J. Del Buono, P. L. Vissat and Frank S. G. Williams, Paper No. 53-PET-31 presented at the Petroleum Division Conference of the ASME, Houston, September 28-30, 1953.

APPENDIX

Bibliography and Abstract of Published Research Work Applicable to Pipe Line Branch Connections

The following is a list of published papers which have come to the authors' attention and which are believed to be particularly applicable to the design of pipe line branch connections.

1. "Effect of Openings in Pressure Vessels" by J. Hall Taylor, and E. O. Waters, presented at the June, 1933 meeting of the ASME. Hydrostatic tests on pressure vessels with 42" inside diameter I.D. x 1" wall thickness with ten openings of various sizes and types of reinforcements. Stresses around the openings was determined by strain gage readings on the outside surface. Results are presented in graphical form.

2. "Investigations of Stress Conditions in a Full-Size Welded Branch Connection" by F. L. Everett and Arthur McCutchan, Transactions of the ASME, July, 1938. Hydrostatic tests of two 8" x 8" pipeline branch connections; one assembly was unreinforced; the other was reinforced with a pad. Strains were measured on the outside surface in the area of the intersection using a Huggenberger mechanical strain gage.

3. "Bursting Pressure Tests of Welded Pipe Headers Show Need for Nozzle Reinforcements" by Eric R. Seabloom, Valve World, July-August, 1941. Hydrostatic tests of 8" x 8" pad reinforced branch connection, 12" x 12" pad reinforced branch connection and a 12" x 6" unreinforced connection. Deformations were measured with indicating dial gages. Pressure was increased until the test assembly fractured.

4. "Design of Fabricated Plate Steel Tees, Laterals and Wyes of Large Diameters for the Pressure Aqueduct of the Boston Metropolitan District Water Supply Commission" by Chester J. Ginder and Edwin B. Cobb, Boston Society of Civil Engineers, Vol. 28, 1941. Describes a method of designing wye branch connections and laterals in the size range of 60 to

108". Details of the designs of several typical branch connections are given.

5. "Strengthening of Circular Holes in Plates Under Edge Loads" by Leon Beskin, Journal of Applied Mechanics, September, 1944. Theoretical stress distributions are determined around strengthened circular holes in plates submitted to edge loads at infinity. Various proportions of circular strengthenings are considered. Theory is approximately applicable to branch connections in which the run pipe is very much larger than the branch pipe so that locally, the run pipe can be considered as a flat plate.

6. "Stresses in a Cylindrical Shell Due to Nozzle or Pipe Connection" by G. J. Schoessow and L. F. Kooistra, Journal of Applied Mechanics, June, 1945. Tests on a 56" diameter shell with 11 3/4" O. D. unreinforced connection. The shell was made in two halves, one half being 1.30" thick the other half 2.08" thick. The branch pipes were 7/8" thick. The shell was anchored while bending or axial loads were applied to the branch pipes. Stresses due to these loadings were measured by means of electrical resistance strain gages.

7. "Reinforcement of Branch Pieces" by J. S. Blair, Engineering (London), July 5, Sept. 6, Nov. 29, Dec. 6, 13, 20, 27, 1946. Hydrostatic, static bending and cyclic bending tests of a large variety of branch connections. The tri-form type of reinforcement is extensively investigated and methods for the design of this type of reinforcement are presented. In the hydrostatic tests, the yield pressure was determined by means of a brittle coating following which the pressure to cause fracture was determined.

8. "Experimental and Analytical Determinations of the Stress Systems in a Welded Pressure Vessel," anonymous, published by British Welding Research Association. Hydrostatic tests of a boiler drum with various sizes of nozzle outlets. Stresses around the nozzle openings are measured by electrical resistance type strain gages. Strains were measured on both the outside and inside surfaces. The tests described in this report were carried out during 1946 and 1947 by Babcock and Wilcox, Ltd.

9. "Analysis of Experimental Data Regarding Certain Design Features of Pressure Vessels" by G. J. Schoessow and E. A. Brooks, Transactions of the ASME, July, 1950. Analysis of data obtained by strain gage experimental stress analysis on drum heads and nozzle openings in heads and shells.

10. "Fatigue Tests of Piping Components" by A. R. C. Markl, Transactions of the ASME, Vol. 74, No. 3. Cyclic bending tests on various pipe components such as reducers, curved pipe, miter bends, forged tees and fabricated branch connections. Results are compared with the strength of straight pipe and generalized formulas for the stress intensification factors of the piping components are presented.

11. "Design of Hot Tap Connections in High Pressure Pipelines" by A. J. Del Buono, P. L. Vissat and Frank G. Williams, presented at the ASME Petroleum Conference, Houston, Texas, September 28-30, 1953. Presents a review of the forces acting on branch connections in a pipeline and discusses the design of such branch connections. Appendix I summarizes some previous research work on branch connections in pressure vessels.

Appendix II is a report on tests of an encirclement tee reinforced branch connection; hydrostatic tests were run on this reinforced branch connection with stresses determined by means of electrical resistance strain gages placed on the outside surface. After completion of the strain gage tests, the pressure was raised until fracture of the specimen occurred.

12. "Deformations and Stresses in Circular Cylindrical Shells Caused by Pipe Attachments" by N. J. Hoff, J. Kempner, S. V. Nardo, and F. V. Pohle, November, 1953, Knolls Atomic Power Laboratory, Report KAPL 921, 922, 923, 924, 925 and 1025. Theoretical investigation of the stresses in a pipeline branch connection under bending loads applied to the attached pipe. The load transmitted by the pipe is assumed to be a load acting on a line segment slightly shorter than the diameter of the branch pipe. A correction for the effects of the finite pipe diameter is obtained in an approximate manner by means of the theory of circular plates.

13. "Researches on Welded Pressure Vessels and Pipelines" by Nicol Gross, British Welding Journal, April, 1954. Hydrostatic tests in which strains were measured on various types of branch connections, both on the outside and inside surfaces. Covers cyclic pressure tests on various types of flat head pressure vessel closures and a forged 6" tee.

14. "Stress Analysis of Split Tee Welded Sleeve Reinforcement of Welding Branch Connection for Hot Tapping," anonymous, Report by The Ladish Company, Metallurgical Department, November 1, 1954. Hydrostatic tests of a 24" x 12" branch connection reinforced with a complete encirclement saddle. Stresses were determined by means of electrical resistance strain gages on both the outside and inside surfaces. Upon completion of the strain gage tests, pressure was increased until fracture of the assembly occurred.

15. "Stresses from Radial Loads in Cylindrical Pressure Vessels" by P. P. Bijlaard, published in Welding Research Supplement of the Welding Journal, December, 1954 and December, 1955. Theoretical development of the stresses in a cylindrical shell resulting from a distributed radial load of rectangular shape and by non-uniform distributed loads giving either circumferential or longitudinal localized bending moments on the shell. Numerical calculations are given in graphical form and the comparison is made with the test data of Schoessow and Kooistra.

16. "Welded Tee Connections" by A. G. Barkow and R. A. Huseby, Welding Research Council Bulletin Series, No. 22, May, 1955. Hydrostatic tests on eight 30" O.D. run x 16" O.D. branch connections. Six of the test assemblies were reinforced with saddles, one with a complete encirclement sleeve, and one with a "Weld-O-let." Stresses were measured on the outside surface by means of electrical resistance strain gages. Upon completion of the strain gage test, pressure was increased until fracture occurred.

17. "Investigation of Static and Fatigue Resistance of Model Pressure Vessels" by Julien Dubuc and Georges Welter, The Welding Journal, July, 1956. Hydrostatic and cyclic pressure tests of 13 1/2" O.D. x 3/4" wall pressure vessels. Vessels had 2" O.D. x 3/8" wall nozzles; initial failures in cyclic pressure tests occurred at these nozzles. Notches and

grooves, machined in the outside surface of the shell to simulate accidental notches that might be present in an actual pressure vessel, proved to be lower stress raisers than the nozzle outlets. Two materials were investigated, one with about 35,000 yield strength, 56,000 ultimate strength; the other with about 77,000 yield strength, 96,000 ultimate strength. Cyclic pressures ranged from zero to approximately the yield pressure of the vessels.

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Proceedings of the American Society of Civil Engineers

FLOW OF NATURAL GAS IN PIPELINES[a]

William T. Ivey,[1] M. ASCE, and James H. Dorough[2]
(Proc. Paper 1194)

SYNOPSIS

Consideration is given to the various flow formulae in use in the design of natural gas transmission pipelines, and to the performance ratings of these lines. Generally, all formulae are derived from the general energy equation, the basic difference being found in the methods of considering the friction coefficient, and in the adjustment for the deviation of gases for supercompressibility. Each formula, after applying all adjustments, assumes the incorporation of an "efficiency factor" to record the deviation of the formula from a selected norm, although no basic norm, or perfect operating condition, has been determined by the industry.

The need for such a norm in the form of a standard formula based on maximum possible efficiency of operation, against which actual line performance may be compared, is emphasized.

The merits of a proposed form for this standard equation is presented herein. Each of the basic differences is examined in order to attempt to simplify the mechanical application of the formula, to permit its more rapid use by manual application and by digital computers, thus permitting a more rapid and thorough examination of the many problems presented by the increasingly complicated pipeline networks of the present day.

HISTORICAL

The history of the development of present day formulae for flow of gas in pipelines had its beginning around the turn of the century. The earlier

Note: Discussion open until August 1, 1957. Paper 1194 is part of the copyrighted Journal of the Pipelines Division of the American Society of Civil Engineers, Vol. 83, No. PL 1, March, 1957.

a. Presented at a meeting of the Pipeline Division, ASCE, February 18, 1957, Jackson, Miss.

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applications did not consider deviation, and generally applied a constant friction coefficient. Such formulae were the formulas of Cox, proposed in 1902, and Rix, proposed in 1904. Other early formulae expressed the friction coefficient as a function of the diameter. Such formulae were those of Oliphant (1904), Unwin (1904), Weymouth (1912), California (1925) and Spitzglass (1926). A third group of formulae expressed the friction coefficient as a function of Reynolds' criterion, a factor considering diameter, velocity, density, and viscosity. Such formulae were: Fritzsche (1908), Lee (1914), White (1929) and McAdams and Sherwood (1926). The Panhandle formulae fall in this category. The Panhandle formulae make the friction coefficient a function of Reynolds' number, with a constant viscosity factor.

The first attempt to produce a standard formula was made in 1935 when T. W. Johnson and W. B. Berwald presented a joint report on flow of natural gas through high-pressure transmission lines, under the sponsorship of the Bureau of Mines and the Natural Gas Department of the American Gas Association, issued as Monograph No. 6 of the Bureau of Mines. Numerous data on gas flow through commercial transmission lines were obtained and analyzed. This was probably the most thorough and comprehensive analysis of flow of gas in pipelines to that date. Johnson and Berwald concluded that for the larger diameter lines, free from condensates and other foreign materials, and operating under steady flow conditions, the metered rates of delivery agreed more closely with the rates calculated from Weymouth's formula than with those calculated from any of the other formulae; and in almost every test, under these conditions, the volume calculated from Weymouth's formula was within a few percent of the metered delivery. Further experiments, under actual conditions of natural gas transmission, were recommended.

The work of Johnson and Berwald was extended in 1956 by R. V. Smith, J. S. Miller, and J. W. Ferguson, in a joint report on flow of natural gas through experimental pipelines and transmission lines. Issued under the sponsorship of the Bureau of Mines and the Pipeline Research Committee of the American Gas Association this publication appeared as Monograph No. 9. The report was intended primarily to present the results of tests conducted on experimental pipelines, and to apply these data to a further improvement of the formula. Tests were made on experimental pipelines constructed of 2, 4, 6 and 8 inch seamless pipe, 8 inch welded pipe, and a specially prepared 3 inch superfinished pipe.

The friction coefficient, "f" was redefined as "resistance coefficient," and a new factor "$\sqrt{1/f}$" termed "transmission factor" was introduced.

Transmission factors for the various experimental pipelines were determined for 2, 4, 6 and 8 inch pipe. Relationships were established between transmission factors and the pertinent variables: Reynolds number, absolute roughness, and internal radius of the pipe. Reynolds numbers for these determinations of transmission factors ranged from a low of 49,900 to 36,000,000. At Reynolds' numbers ranging from 250,000 to approximately 850,000 transmission factors for the two inch seamless were found to be related to Reynolds' numbers, internal radius of the pipe, and the absolute roughness of the internal surface of the pipe. At Reynolds numbers above 850,000 the transmission factors were constant and independent of the Reynolds' number, being influenced only by the absolute roughness and the internal radius of the pipe. For larger diameters, extrapolated data indicated the break between transition and fully rough flow moved to higher Reynolds' numbers.

Analysis of pipeline flow efficiency field test data by the Institute of Gas Technology in 1952 and 1953 disclosed inconsistencies in the data resulting from differences in measuring technique. To satisfy the need for proper evaluation of field measurement procedures, and for development of criteria for judging the validity of field data, a set of preliminary recommendations for efficiency testing was developed in 1954. In May, 1956, a formal procedure for pipeline efficiency testing was presented by R. F. Bukacek and R. T. Ellington.

Currently the American Gas Association, working with the Institute of Gas Technology at Chicago, through its Project NB-13, "Pipeline Efficiency Flow Standards" is in the process of analyzing an appreciable quantity of test data. These data, provided through the cooperation of a number of pipeline companies, cover field tests of pipeline installations and include a wide range of diameters and various flow rates. No conclusions have, as yet, been published by this committee.

It is not proposed here to anticipate the conclusions of this committee. It is desired to consider certain data, consisting of over two hundred tests made on pipelines of Southern Natural Gas Company, and to present certain conclusions reached upon examination of that data. The tests were made under the supervision of the Operating Department of Southern Natural Gas Company.

The Basic Formula

Much information is available in the literature on flow of gas in pipelines. Numerous varying formulae are offered for the practical solution of the problem. They all originate from the basic flow equation, as derived from the general energy equation. The basic derivation of the general flow formula appears in the work of Johnson and Berwald in U. S. Bureau of Mines Monograph No. 6 and will not be repeated here.

The general flow formula:

$$Q = K \frac{T_o}{P_o} \left[\frac{(P_1^2 - P_2^2)\, d^5}{GTLf} \right]^{1/2} \qquad (1)$$

where

- Q = volume of gas in cubic feet per hour at pressure base P_o and temperature base T_o;
- K = numerical constant 1.6156;
- T_o = temperature base, defining one cubic foot of gas, degrees F absolute;
- P_o = pressure base defining one cubic foot of gas, pounds per square inch absolute;
- P_1 = inlet pressure, pounds per square inch, absolute;
- P_2 = outlet pressure, pounds per square inch, absolute;
- d = internal diameter of pipe, inches;
- G = specific gravity of gas (air = 1.000);
- T = temperature of flowing gas, degrees F absolute;
- L = length of pipe in miles;
- f = coefficient of friction.

This gives no recognition to the deviation factor, which is recognized by Smith, Miller, and Ferguson in Monograph No. 9 as follows:

$$Q = K \frac{T_o}{P_o} \sqrt{1/f} \sqrt{1/Z} \left[\frac{(P_1^2 - P_2^2)\, d^5}{GTL} \right]^{1/2} \qquad (2)$$

where

f = resistance coefficient, dimensionless;
$\sqrt{1/f}$ = transmission factor, dimensionless;
Z = average effective compressibility factor of gas, dimensionless;

It is noted that "f," formerly defined as "coefficient of friction," is now defined "resistance coefficient," and a new term " $\sqrt{1/f}$," defined as "transmission factor" is introduced. This change was made in an effort to establish clear and concise terminology for these terms.

This formula is the basic flow equation from which all other formulae are derived. At this point, theory gives way to laboratory or field test in the evaluation of the "f" term, or the term that will bring the equation in balance. Three basic approaches have been used in the determination of the various formulae proposed: (1) the application of the friction factor, or resistance coefficient as a constant; (2) its application as a function of the diameter of the pipe only; and (3) as a function of the Reynolds number.

Reynolds Number

The Reynolds' criterion, a dimensionless ratio commonly used to characterize the conditions of fluid flow in circular pipes, is stated as:

$$R = \frac{DUS}{V} \qquad (3)$$

where:

R = Reynolds' criterion
D = Internal diameter of pipe
U = Average linear velocity
S = Density of fluid
V = Viscosity

Reynolds observed that at low velocities flow proceeded in a steady straight-line manner, in which it appeared that all of the particles moved parallel to the walls of the tube. As velocities increased, a certain critical change in the character of the flow occurred, and turbulence appeared. Smith, Miller and Ferguson noted a change in the relationship between the transmission factor and Reynolds' number as this critical point was reached. They stated that transmission factor variations, for the experimental data on two inch diameter pipe, depended on Reynolds' numbers only for ranges in Reynolds' numbers from 49,900 to 250,000. At Reynolds numbers ranging from 250,000 to approximately 850,000 transmission factors were related to Reynolds' numbers, internal radius of the pipe, and the absolute roughness of the internal surface of the pipe. At Reynolds' numbers above 850,000, for the

pipes considered, transmission factors were constant and independent of Reynolds' numbers. They were influenced only by the absolute roughness and the internal diameter of the pipe. For larger diameters, extrapolated data indicated the break between transition and fully rough flow moved to higher Reynolds' numbers. Smith, Miller and Ferguson interpreted these three areas of flow as "smooth," "transition," and "fully rough," which compare with Reynolds observations of "steady" and "turbulent" areas of flow.

Consideration of Test Data

Over the past several years, a continuing series of careful tests have been carried out on the pipelines of Southern Natural Gas Company under the supervision of the Operating Department of Southern Natural Gas Company. Pipelines tested consisted of sizes ranging from ten-inch to twenty-four inch, inclusive. Flow measurement was made by metering where possible, and checked by means of ammonia tracer. Due to the variable distances of orifice measurement from the point of test, the ammonia tracer method was found to give more consistent results. These tests are summarized in Appendix A.

Volumes of gas are stated in Mcf per day at a pressure base of 14.73 psia and are as measured by ammonia tracer. Where ammonia tracer measurement was not available, metered flow is used and the values so obtained are set out by an asterisk. The table shows the measured flow, the internal diameter of the pipeline in inches, the difference between the squares of the initial and terminal absolute pressures, the average absolute pressure in the test section, the gravity of the gas, the absolute temperature, the deviation factor and the gas viscosity. An adjustment is made for elevation. This adjustment is discussed in Appendix B.

A solution is made for Reynolds' number, resistance coefficient, and transmission factor.

Certain conclusions are immediately evident:

1. Two hundred and thirty-three tests were conducted in actual operation of a large transmission system. Under varying ranges of normal operating conditions, results were obtained from varying lengths of pipe ranging in nominal size from ten to twenty-four inches and under pressures ranging from three hundred to twelve hundred pounds per square inch gage. During this entire series Reynolds' numbers varied from a low of 2,202,511 for one twelve inch pipe to a high of 13,272,727 for one eighteen inch pipe. That is, all of the tests made showed Reynolds' numbers in ranges much higher than Reynolds' numbers in the "smooth" and "transition" ranges defined by Smith, Miller, and Ferguson. They were also well above the Reynolds' number which, according to Smith, Miller and Ferguson, define the limit above which the transmission factors are influenced only by the absolute roughness and internal diameter.

   Therefore, if this conclusion can be borne out by further tests, normal pipeline operation need not be concerned with the Reynolds' criterion, but only by diameter and by absolute roughness.

2. Plots made of transmission factor against Reynolds' number for 16, 18 and 24 inch diameters from the test data available show a broad band of points at each Reynolds number and no clearly defined relationship between

Reynolds' number and transmission factor within the range of the tests made.

If the influence of Reynolds' number is not felt in rough, turbulent flow, but rather if the transmission factor is influenced only by the absolute roughness and internal diameter, then the data should be examined by sizes of pipe, and roughness factors calculated for each pipe size. The formula used by Smith, Miller and Ferguson is their resistance equation (Formula 28, Monograph 9):

$$\sqrt{1/f} = 4 \log. \frac{7.4\ r}{k} \qquad (4)$$

where r = internal radius of pipe in inches
k = absolute roughness coefficient

The results of this examination are tabulated in Table I.

Each line in Table I represents a group of tests on a single line or lines of the same diameter and in the same continuing run of pipeline. The nominal pipe size is shown in the first column as a matter of convenience, but all calculations are based on the internal diameter shown in the second column. The number of tests made, and the maximum, minimum and average transmission factor are shown for each group of tests. The absolute roughness factor is calculated by the equation (4), on the assumption that all Reynolds numbers exceed the critical criteria for rough turbulent flow, and therefore, the transmission factor is a function of diameter and absolute roughness only.

Line 1, Table I, gives the results of a total of twenty tests on a new solid welded twenty-four inch pipeline operating in the range of 500 to 750 pounds per square inch gage. The transmission factor varies from 19.741 to 20.933 with an average transmission factor of 20.195, or for a deviation of less than 3%, plus or minus. The calculated absolute roughness coefficient is 0.000777. This line is one of the most recent installations and has been in operation for the entire period on clean gas. Smith, Miller and Ferguson reported absolute roughness from 0.000553 to 0.00185, and used an absolute roughness factor of 0.0007 in the calculation of the limits of the transition region.

For the purpose of this analysis, it may be assumed that an absolute roughness of 0.000777 and a transmission factor of 20.000 for 24 inch pipe indicates a line operating at high efficiency.

Line 2 gives the results of eleven tests on twenty-two inch pipe and shows an absolute roughness factor of 0.000721. Line 5 gives the results of 18 tests on twenty-two inch pipe with an absolute roughness of 0.000624. Line 17 gives the results of four tests on eighteen inch pipe with an absolute roughness of 0.000691. Line 23 gives five tests on sixteen inch with an absolute roughness of 0.000792 and line 24 gives five tests on eighteen inch with 0.000789. Line 33 gives 17 tests on 18 inch with 0.000761. All of these tests may be interpreted as tests on pipe in good condition, operating efficiently.

Line 3 gives the results of eight tests on twenty-inch pipeline with an absolute roughness of 0.002596. Line 4 gives the results of six tests on twenty-four inch with an absolute roughness of 0.001503. Line 7 gives four tests on twelve-inch with 0.000848 coefficient, line 8 gives sixteen tests on

TABLE I
COMPARISON OF TRANSMISSION FACTORS AND ABSOLUTE ROUGHNESS COEFFICIENTS BY PIPE SIZES FROM TESTS MADE BY SOUTHERN NATURAL GAS COMPANY

Line	PIPE SIZE INCHES O.D.	PIPE SIZE INCHES I.D.	Number of Tests	TRANSMISSION FACTORS Minimum	TRANSMISSION FACTORS Maximum	TRANSMISSION FACTORS Average	Absolute Roughness Coefficient
1	24	23.500	20	19.741	20.933	20.195	0.000777
2	22	21.375	11	19.630	20.943	20.160	0.000721
3	20	19.313	8	16.265	18.818	17.591	0.002596
4	24	23.188	6	17.565	19.803	19.026	0.001503
5	22	21.375	18	19.245	21.320	20.413	0.000624
6	18	17.375	22	18.315	20.528	19.350	0.000935
7	12 3/4	12.125	4	18.257	19.518	18.893	0.000848
8	20	19.313	16	17.678	19.952	18.921	0.001329
9	24	23.188	6	16.860	20.240	19.207	0.001354
10	14	13.375	1			15.811	0.005518
11	16	15.375	1			18.257	0.001552
12	12 3/4	12.125	1			16.514	0.003372
13	20	19.313	9	16.566	19.642	18.846	0.001389
14	20	19.313	2	18.474	20.228	19.351	0.001038
15	14	13.375	13	17.413	19.668	18.752	0.001015
16	16	15.438	7	18.788	19.691	19.400	0.000807
17	18	17.375	4	19.830	19.940	19.875	0.000691
18	16	15.438	1			19.245	0.000882
19	18	17.375	4	19.181	19.462	19.355	0.000932
20	16	15.375	9	18.380	20.000	19.211	0.000950
21	12 3/4	12.250	3	18.898	20.000	19.503	0.000603
22	10 3/4	10.250	4	18.257	20.000	18.896	0.000716
23	16	15.438	5	18.945	19.745	19.432	0.000792
24	18	17.375	5	18.709	20.024	19.645	0.000789
25	20	19.250	2	17.140	17.321	17.230	0.003509
26	22	21.375	10	16.391	21.063	19.288	0.001191
27	18	17.500	2	18.098	18.257	18.178	0.001848
28	22	21.375	4	17.874	19.563	18.709	0.001663
29	18	17.375	4	18.411	19.121	18.768	0.001307
30	22	21.375	1			19.415	0.001075
31	24	23.390	1			20.000	0.000865
32	24	23.500	4	18.452	18.966	18.688	0.001850
33	18	17.375	17	18.676	20.357	19.706	0.000761
34	20	19.313	8	17.639	19.877	18.620	0.001581

twenty-inch with 0.001329, and line 9 gives six tests on twenty-four inch with 0.001354. All of these lines are known to have contained foreign matter, presumably liquids, at the time of the tests.

Lines 10, 11 and 12 were tests on field gathering lines which from time to time had received liquids. These tests may be presumed to show results on lines which have operated at poor efficiency.

Lines 25, 26, 27, 28, 29 and 30 record tests on the oldest portion of the system, a great part of which consisted of coupled lines. These tests may be presumed to show results on old lines which have operated for a number of years.

Thus, a range of absolute roughness factors may be determined to show operating conditions from maximum efficiency to poor efficiency.

It is concluded that, for the tests considered, the condition of the pipeline, with respect to liquids or other foreign matter, and with respect to the age of the pipeline and type of construction, inclusion of couplings, valves, and cross connections, has a much greater influence on the transmission factor than the pipe diameter or Reynolds' criteria.

The Dilemma of the Gas Industry

It is apparent from the above discussion that it is not possible to specifically express the percent efficiency of a pipeline without a basic norm or standard base of reference. Each gas transmission company rates its pipelines according to its own formula, assuming friction coefficients based on either Reynolds' numbers or pipe diameters, and on roughness coefficients related or not related to diameters. A line that may be rated at high efficiency by one formula will be rated at a low efficiency by another. In fact, by some formulae, a line may be rated at over 100% efficiency.

It is evident that line efficiency cannot be adequately determined by friction coefficients based on a constant without reference to line size. It also appears that in many actual operating conditions the friction coefficient or transmission factor is independent of Reynolds' number. May it be assumed that a practical solution for commercial pipelines under normal operating conditions is in the area of high Reynolds' numbers with transmission factors based on diameters and absolute roughness alone?

Smith, Miller and Ferguson suggest a tentative average absolute roughness of 0.0007, and agree that for fully rough flow the transmission factor is not a function of Reynolds number. On this basis, transmission factors are calculated for varying pipe diameters in Table II.

Such transmission factors are not offered as final, but are suggested as probably representative of transmission factors for turbulent flow in areas of high Reynolds' numbers. Acceptance of a basic absolute roughness coefficient for pipe of maximum practical smoothness will permit the determination of transmission factors which might be used as standards of maximum line efficiency.

The Deviation Factor

It has been shown that natural gas does not follow the ideal gas relationship, but that, at high pressures greater volumes of gas may be compressed into a given space than the ideal gas laws indicate. Failure to recognize this

TABLE II
TRANSMISSION FACTORS FOR ROUGH TURBULENT FLOW
THROUGH PIPELINES OF VARIOUS DIAMETERS

$$\left[\sqrt{1/f} = 4 \log d + 14.9; \quad K = 0.0007\right]$$

Pipe Size: Inches O.D.		Wall	I.D.	Transmission Factor
4 1/2	x	3/16	4.125	17.362
6 5/8	x	1/4	6.125	18.048
8 5/8	x	1/4	8.125	18.539
10 3/4	x	1/4	10.250	18.943
12 3/4	x	1/4	12.250	19.253
14	x	1/4	13.500	19.421
16	x	5/16	15.375	19.647
18	x	5/16	17.375	19.860
20	x	11/32	19.3125	20.043
22	x	3/8	21.250	20.209
24	x	13/32	23.1875	20.361
26	x	7/16	25.125	20.500
30	x	7/16	29.125	20.757

deviation in the past has not introduced a material error because of the low pressures at which earlier pipelines operated. However, with continuing increases in operating pressures, recognition of this deviation or supercompressibility characteristic of the gas becomes more and more important.

This subject has been most thoroughly covered by Dr. George Granger Brown in his paper, "Deviation of Natural Gas from Ideal Gas Laws." Dr. Brown points out that although numerous equations of state have been proposed, the most convenient method for computing the pressure-volume-temperature relationship of natural gas is by the use of the so-called compressibility factor, which represents the deviation of the gas in question from the ideal gas laws, as expressed by the equation

$$PV = ZNRT \tag{5}$$

where

P = pressure in pounds per square inch absolute
V = Volume in cubic feet
Z = the compressibility factor
N = number of pounds mol
R = 10.71 for all gases
T = absolute temperature

The compressibility factor Z is a dimensionless intensive factor

independent of the extent or weight of the gas, and determined by the character of the gas, the temperature, and the pressure. Once Z is known or determined, the calculation of the pressure-temperature-volume relationship may be made with as much ease at high pressure as at low pressure.

According to the theorem of corresponding states, the deviation of an actual gas from the ideal gas law is the same for different gases when at the same corresponding state. The same corresponding states are found at the same fraction of the absolute critical temperature and pressure which are known respectively as

$$\text{Reduced Temperature } T_r = \frac{T}{T_c} \tag{6}$$

$$\text{Reduced pressure } P_r = \frac{P}{P_c} \tag{7}$$

where

T_c = the absolute critical temperature
P_c = the absolute critical pressure
T = the absolute temperature at which the gas exists
P = the absolute pressure at which the gas exists
T_r = reduced temperature
P_r = reduced pressure

Critical temperature is defined to be the temperature above which a pure gas cannot be liquefied by pressure alone. Critical pressure is defined to be the pressure below which a pure gas may exist in a gaseous state in equilibrium with its liquid.

Therefore, if the theorem of corresponding states can be applied without appreciable error, all gases would have the same value of Z at the same reduced temperature and pressure, and a plot of Z for methane, as a function of reduced temperature and pressure, can be applied to determine the unknown value of Z for some other gas if we know or can determine the critical temperature and pressure of the second gas.

In the previously mentioned paper Dr. Brown presented such a plot for methane and for natural gas which has found wide use in the industry in the determination of compressibility factors.

Based on this chart, deviation factors are determined for each of the various gas mixtures found in the system on which gas tests were made. Characteristics of the gas considered are determined by laboratory analysis. Individual absolute critical temperatures and pressures of each constituent are set down and the pseudo-reduced temperature for the mol-fraction calculated. The arithmetic total of these fractions will give the absolute pseudo-critical temperature and absolute pseudo-critical pressure for the gas mixture considered. From these values the pseudo-reduced temperatures and pseudo-reduced pressures for varying actual temperatures and pressures are determined. From these factors, the deviation factors may be read from the chart.

As Dr. Brown points out, like all correction factors, the value of Z depends upon experimental data for its determination.

The standard apparatus for experimentally determining supercompressibility is either the Bean or the Burnette. The Bean apparatus consists of a high pressure gas bomb of known volume and a calibrated burette for measuring the volume of the gas at atmospheric pressure. It is simply an adaptation

of the classic laboratory procedure of expanding an unknown quantity of gas in successive increments from a high pressure bomb into a calibrated burette at essentially atmospheric pressure, and computing the total volume from a summation of these increments. The operation is tedious; mistakes are easy to make, and one mistake can ruin the whole run. The Burnette apparatus consists of two adjacent high pressure chambers of a known volume ratio, the gas being expanded from the first chamber into the evacuated second chamber, with successive expansions being made after re-evacuation of the second chamber. The deviation of the gas is computed from a comparison of the volume ratio of the chambers to the pressure ratio before and after the expansion. The operation is very easy and not conducive to mistakes. The methods are of comparative accuracy, agreeing to within about 0.1%. However, due to ease of operation, the Burnette is most used.

In 1955, the American Gas Association, through PAR Project NX-7, prepared such data based on the specific gravity method of super-compressibility factor determination. Tables of supercompressibility factors F_{pv} for natural gas were issued for nominal temperatures from 0° to 180° F, for specific gravity from 0.554 to 0.750, and for pressures of from 0 to 3,000 psig. Separate tables were included for the determination of supercompressibility factors for natural gas containing nitrogen and or carbon dioxide.

By definition

$$F_{pv} = \sqrt{\frac{NRT}{PV}} \tag{8}$$

From Equation 5

$$\sqrt{\frac{1}{Z}} = \sqrt{\frac{NRT}{PV}} \tag{9}$$

Therefore,

$$F_{pv} = \sqrt{\frac{1}{Z}} \tag{10}$$

In Table III, a comparison is made between Z factors calculated by the Granger Brown method and F_{pv} factors as taken from the A.G.A. tables. The differences are negligible, and it may be concluded that deviation factors may be determined by either method.

A Proposed Formula for Flow of Natural Gas

It appears, at least for the data available here, that the following conclusions can be drawn:

1. For turbulent flow the transmission factor is not a function of Reynolds number
2. Normal commercial operation will fall within this range
3. The transmission factor will be a function of diameters and absolute roughness
4. The absolute roughness coefficient for pipelines in good condition should approximate 0.0007
5. The deviation factor may be obtained by the George Granger Brown method or from the A.G.A. tables of deviation
6. All of these factors may readily be incorporated in the basic flow equation.

TABLE III

COMPARISON OF DEVIATION FACTOR Z OF WESTFIELD GAS CALCULATED BY GRANGER BROWN METHOD WITH F_{pv} FROM A.G.A. TABLES

(1) Pressure Psig	(2) "Z" at 60°F	(3) $\frac{1}{Z}$ at 60°F	(4) $\sqrt{\frac{1}{Z}}$	(5) A.G.A Tables F_{pv}
100	0.983	1.0173	1.0086	1.0083
200	0.967	1.0341	1.0169	1.0164
300	0.950	1.0526	1.0260	1.0246
400	0.934	1.0707	1.0347	1.0329
500	0.918	1.0893	1.0437	1.0416
600	0.903	1.1074	1.0523	1.0501
700	0.890	1.1236	1.0600	1.0589
800	0.876	1.1416	1.0685	1.0677
900	0.862	1.1601	1.0771	1.0763
1000	0.848	1.1792	1.0859	1.0847
1100	0.837	1.1947	1.0930	1.0933
1200	0.825	1.2120	1.1009	1.1015

Therefore, for a design formula applicable to normal commercial use of pipelines in good condition, the basic flow equation may be used as follows:

Proposed Formula

$$Q = K \frac{T_o}{P_o} \sqrt{1/f} \quad \sqrt{1/Z} \left[\frac{(P_1^2 - P_2^2)\, d^5}{GTL} \right]^{1/2} \qquad (11)$$

For normal use, a constant (K_1) may be determined to include all constant factors, including diameter and transmission factor and a table prepared to give K_1 for each diameter. The equation then becomes:

$$Q = K_1 \left[\frac{P_1^2 - P_2^2}{L} \right]^{1/2} \times \sqrt{1/Z} \qquad (12)$$

At this point consideration is given to the application of the factor $\sqrt{1/Z}$. In most applications, this is taken to be the deviation for the gas at the mean effective pressure between the initial and terminal pressure.

A compressibility factor may be determined, as more fully outlined in Appendix B, which may be applied directly to the pressure.

The modified flow formula then becomes:

$$Q = K_1 \left[\frac{(P_1 \cdot b_1)^2 - (P_2 \cdot b_2)^2}{L} \right]^{1/2} \qquad (13)$$

A plot of deviation factors for a representative gas may then be determined for varying pressures, and a tabulation prepared adjusting the actual pressure to an adjusted pressure recognizing the deviation factor.

In this form, through the use of a table of constants taking into account diameter and transmission factor for each pipe size, another a table giving squares of absolute pressures, adjusted for the deviation of a typical gas for varying gage pressures, the formula is readily applied without the use of graphs, charts, logarithms, and without cut and try. It may be handled rapidly in manual operation, with the use of a desk calculator, or may be keyed into the operation of any of the basic digital computers.

In this manner, through the use of digital computers, it will be possible to make a much more rapid and more thorough examination of the many and complicated problems both of construction and of operation facing the increasingly complicated pipeline networks of today.

CONCLUSIONS

Many different formulae have been presented for determination of flow of gas in pipelines. All are derived from the basic flow equation. Chief difference is in the determination of friction coefficient. Some formulae set a basic friction factor for all conditions; some relate friction factor or transmission factor to diameter and absolute roughness. Others relate transmission factor to Reynolds number. Early formulae did not adjust for deviation. It is generally accepted today that such adjustment is necessary.

For practical application, based on the tests submitted herein, it is reasonable to conclude transmission factor is not a function of Reynolds' number for turbulent flow in the area of normal commercial operation, but that transmission factor is a function of diameter and absolute roughness. It appears that variations of absolute roughness due to varying pipe conditions is of greater significance than pipe diameter. It is urged that a basic absolute roughness factor, such as 0.0007, be agreed upon as being the norm against which pipeline efficiency can be determined.

TABLE IV – APPENDIX A

SOUTHERN NATURAL GAS COMPANY PIPELINE TESTING PROGRAM
CALCULATION OF REYNOLDS NUMBER, RESISTANCE COEFFICIENT AND TRANSMISSION FACTOR

Test No.	Mcfd @ 14.73#	Internal Dia. Inch.	$P_1^2 - P_2^2$	P Avg.	Elevation Factor	Grav.	Temp. Abs.	Dev.	Length Pipe Mi.	Gas Visc.	Reynolds Number	Resist. Coef.	Trans. Fact.
1	183,332*	23.500	19,934	717	3,896-	.594	518	.890	9.897	7.78	8,060,056	.002363	20.572
2	174,484	23.500	19,111	703	568-	.594	518	.895	11.491	7.75	7,700,598	.002594	19.634
3	174,114	23.500	21,984	687	1,242-	.594	519	.897	12.084	7.72	7,714,309	.002762	19.028
4	181,362	23.500	20,917	670	1,291-	.594	518	.900	11.655	7.70	8,056,314	.002481	20.076
5	173,068	23.500	17,852	663	1,473-	.594	518	.901	10.314	7.68	7,707,890	.002558	19.772
6	181,160	23.500	13,135	650	4,315+	.594	518	.903	11.111	7.67	8,078,613	.002317	20.775
7	177,013	23.500	12,769	726	3,843+	.594	527	.898	10.594	8.00	7,568,245	.002398	20.421
8	177,670	23.500	14,112	720	1,447-	.594	524	.897	7.924	7.95	7,643,901	.002429	20.290
9	177,013	23.500	17,836	708	2,477+	.594	522	.897	12.561	7.88	7,683,497	.002505	19.980
10	184,761	23.500	26,488	692	1,392-	.594	521	.899	14.935	7.80	8,102,073	.002373	20.528
11	178,277	23.500	29,117	672	4,462-	.594	522	.903	14.517	7.75	7,867,951	.002566	19.741
12	180,249	23.500	16,694	653	1,661+	.594	523	.906	11.061	7.71	7,996,247	.002436	20.261
13	177,417	23.500	11,559	643	2,238+	.594	523	.907	8.653	7.70	7,881,072	.002403	20.400
14	179,187	23.500	18,365	630	1,472+	.594	523	.900	11.801	7.69	7,969,843	.002509	19.964
15	183,891	23.500	23,139	612	2,489-	.594	523	.906	12.424	7.68	8,189,938	.002347	20.641
16	178,934	23.500	26,922	593	6,170-	.594	522	.914	13.032	7.66	7,989,945	.002364	20.567
17	182,475	23.500	23,962	569	2,054+	.594	521	.916	15.620	7.62	8,190,819	.002374	20.524
18	182,879	23.500	22,106	550	1,956-	.594	522	.919	11.926	7.58	8,252,288	.002377	20.511
19	183,183	23.500	16,784	532	699+	.594	522	.923	10.698	7.57	8,276,728	.002282	20.933
20	178,631	23.500	15,373	516	619+	.594	522	.924	8.778	7.55	8,092,374	.002688	19.288

Test No.	Mcfd @ 14.73#	Internal Dia. Inch.	$P_1^2 - P_2^2$	P Avg.	Elevation Factor	Grav.	Temp. Abs.	Dev.	Length Pipe Mi.	Gas Visc.	Reynolds Number	Resist. Coef.	Trans. Fact
21	166,948	21.375	27,590	448	1,477-	.599	519	.931	10.389	7.47	8,475,105	.002595	19.630
22	164,268	21.375	26,532	415	191-	.599	527	.939	11.465	7.56	8,239,480	.002391	20.451
23	164,571	21.375	28,408	380	364-	.599	526	.947	12.186	7.50	8,321,128	.002388	20.464
24	163,206	21.375	26,475	343	321-	.598	523	.951	11.535	7.38	8,371,981	.002407	20.383
25	161,688	21.375	24,029	304	261-	.598	524	.956	10.279	7.30	8,385,029	.002488	20.048
26	161,638	21.375	25,009	451	1,878+	.598	541	.941	11.063	8.05	7,601,558	.002557	19.776
27	163,406	21.375	24,686	424	1,246+	.599	533	.943	10.583	7.67	8,078,744	.002553	19.791
28	161,413	21.375	19,360	397	418-	.599	529	.945	7.926	7.57	8,085,780	.002548	19.811
29	163,047	21.375	26,845	366	628+	.599	526	.947	12.449	7.46	8,287,972	.002323	20.748
30	167,563	21.375	34,378	321	280-	.596	525	.954	14.873	7.40	8,543,495	.002280	20.943
31	160,237	21.375	16,619	279	41-	.596	525	.960	6.955	7.33	8,248,149	.002574	19.710
32	107,931*	19.312	37,586	895		.612	523	.864	15.631	8.25	5,610,344	.003780	16.265
33	109,436*	19.312	28,702	872		.610	528	.873	15.094	8.33	5,615,342	.002824	18.818
34	106,620*	19.313	25,638	846		.618	531	.874	12.638	8.30	5,562,321	.003146	17.829
35	105,514	19.313	23,962	831		.616	531	.880	10.124	8.20	5,553,703	.003781	16.263
36	105,190	19.313	21,807	818	932-	.608	532	.883	11.651	8.20	5,464,734	.002838	18.771
37	104,634	19.313	23,504	804	991-	.608	533	.887	12.279	8.18	5,449,234	.002897	18.579
38	104,108	19.313	31,731	775	3,499-	.608	534	.891	12.317	8.20	5,408,474	.003641	16.572
39	106,414	19.313	22,478	759	1,181+	.608	535	.893	11.291	8.19	5,535,247	.003216	17.633
40	145,409	23.188	15,249	748	2,414-	.608	534	.894	9.918	8.17	6,314,806	.002639	19.466
41	148,747	23.188	15,305	743	2,266-	.608	533	.895	9.524	8.13	6,491,672	.002689	19.284
42	160,734	23.188	21,622	731	1,246-	.608	532	.897	10.492	8.09	7,049,576	.003241	17.565
43	159,570	23.188	17,906	716	5,206+	.608	533	.898	14.043	8.00	7,077,358	.002788	18.939

Test No.	Mcfd @ 14.73#	Internal Dia. Inch.	$P_1^2 - P_2^2$	P Avg.	Elevation Factor	Grav.	Temp. Abs.	Dev.	Length Pipe Mi.	Gas Visc.	Reynolds Number	Resist. Coef.	Trans. Fact.
44	160,531	23.188	29,684	700	7,146-	.603	532	.903	14.846	7.91	7,141,533	.002550	19.803
45	159,216	23.188	15,026	683	596-	.603	532	.904	9.037	7.81	7,173,771	.002742	19.097
46	181,321	21.375	30,262	477	1,699-	.589	530	.935	9.954	7.70	8,780,546	.002490	20.040
47	182,839	21.375	32,411	442	216-	.589	536	.943	11.293	7.72	8,830,990	.002402	20.404
48	182,697	21.375	34,439	402	409-	.589	532	.947	11.886	7.66	8,893,666	.002407	20.383
49	178,647*	21.375	26,791	362	360-	.589	530	.950	9.203	7.57	8,799,703	.002540	19.842
50	185,681	21.375	17,615	333	318-	.587	530	.956	6.198	7.54	9,151,331	.002266	21.007
51	184,169	21.375	30,345	495	2,297+	.587	544	.940	10.897	7.94	8,619,491	.002459	20.166
52	184,852	21.375	28,797	464	1,531+	.587	539	.943	10.424	7.81	8,795,555	.002387	20.468
53	185,893	21.375	22,193	437	518-	.587	535	.943	7.805	7.71	8,959,951	.002263	21.021
54	185,428	21.375	19,834	412	814+	.587	532	.947	7.507	7.64	9,019,105	.002276	20.961
55	179,691*	21.375	43,108	364	367-	.586	531	.953	14.446	7.57	8,806,069	.002587	19.661
56	190,514	21.375	11,926	325	58-	.586	530	.956	4.194	7.50	9,423,804	.002200	21.320
57	181,564	21.375	24,076	500	2,177-	.586	544	.940	7.547	7.94	8,483,149	.002440	20.244
58	186,778	21.375	31,308	472	702+	.586	536	.941	11.043	7.76	8,929,282	.002351	20.624
59	181,756	21.375	22,258	441	1,016+	.588	531	.941	8.664	7.63	8,867,496	.002310	20.806
60	182,313	21.375	31,479	410	606+	.588	531	.945	11.876	7.58	8,953,401	.002291	20.892
61	177,164	21.375	23,438	375	882-	.588	530	.948	7.537	7.53	8,758,372	.002700	19.245
62	181,227	21.375	23,672	341	1,984-	.593	529	.953	7.854	7.48	9,095,316	.002351	20.624
63	178,113*	21.375	33,020	296	533+	.593	528	.958	11.466	7.43	8,999,307	.002569	19.729
64	239,523	17.375	98,542	1,193	4,575+	.587	552	.872	7.172	10.90	10,045,937	.002638	19.470
65	231,330	17.375	119,260	1,146	2,004-	.587	549	.874	8.872	10.65	9,930,393	.002597	19.623
66	220,517	17.375	96,678	1,095	815-	.587	544	.874	7.741	10.10	9,981,366	.002705	19.227

Test No.	Mcfd @ 14.73#	Internal Dia. Inch	$P_1^2 - P_2^2$	P Avg.	Elevation Factor	Grav.	Temp. Abs.	Dev.	Length Pipe Mi.	Gas Visc.	Reynolds Number	Resist. Coef.	Trans. Fact.
67	232,402	17.375	102,610	1,050	9,747+	.587	535	.870	8.713	9.30	11,424,160	.002581	19.683
68	232,392	17.375	119,899	995	456-	.585	536	.878	8.318	9.05	11,699,758	.002855	18.715
69	231,158	17.375	110,088	938	1,553+	.585	535	.882	8.106	8.80	11,967,953	.002758	19.042
70	229,176	17.375	105,234	878	641-	.585	534	.889	7.477	8.55	12,211,968	.002844	18.751
71	227,102	17.375	148,308	804	3,099+	.585	534	.899	10.704	8.30	12,466,473	.002897	18.579
72	241,789*	17.375	134,232	1,166	13,525-	.585	543	.865	8.828	10.50	10,491,449	.002521	19.916
73	230,147	17.375	171,754	1,100	1,193+	.588	541	.869	13.724	10.15	10,383,534	.002552	19.795
74	229,702	17.375	97,361	1,035	6,154+	.588	538	.872	7.694	9.35	11,250,154	.002741	19.101
75	230,723	17.375	100,385	982	6,276+	.588	537	.878	7.896	9.13	11,572,969	.002701	19.241
76	229,499	17.375	103,378	928	2,967+	.586	535	.883	7.916	8.75	11,970,466	.002739	19.107
77	227,345	17.375	119,224	864	861+	.587	535	.891	9.003	8.50	12,227,368	.002738	19.111
78	234,537	17.375	137,742	782	2,730-	.587	532	.899	10.926	8.21	13,059,867	.002373	20.528
79	233,798	17.375	74,719	709	46-	.587	530	.906	5.638	8.05	13,277,472	.002552	19.795
80	170,053*	17.375	79,992	1,148	2,854-	.588	542	.864	11.890	10.20	7,634,599	.002411	20.366
81	161,562	17.375	47,554	1,119	2,144+	.588	538	.863	8.271	9.87	7,496,181	.002492	20.032
82	158,340	17.375	112,689	1,084	4,142-	.588	536	.864	15.863	9.54	7,600,627	.002981	18.315
83	163,701	17.375	83,697	1,039	6,026+	.599	536	.864	12.301	9.37	8,150,491	.002923	18.496
84	165,916	17.375	134,765	983	14,069-	.599	535	.869	17.320	9.00	8,599,949	.002692	19.274
85	165,775	17.375	105,347	920	11,401+	.599	536	.878	17.019	8.79	8,798,010	.002621	19.533
86	53,752*	12.125	56,946	950		.598	534	.873	12.574	8.79	4,081,160	.002800	18.898
87	53,752*	12.125	29,372	927		.598	531	.872	6.392	8.61	4,166,379	.002800	18.898
88	45,982	12.125	43,062	906		.614	537	.874	12.574	8.69	3,625,795	.002625	19.518
89	46,659	12.125	24,805	890		.614	536	.876	6.490	8.63	3,704,797	.003000	18.257

Test No.	Mcfd @ 14.73#	Internal Dia. Inch.	$P_1^2 - P_2^2$	P Avg.	Elevation Factor	Grav.	Temp. Abs.	Dev.	Length Pipe Mi.	Gas Visc.	Reynolds Number	Resist. Coef.	Trans. Fact.
90	114,836	19.313	35,197	898		.599	536	.882	15.631	8.64	5,578,449	.003052	18.101
91	121,661*	19.313	33,931	884		.599	537	.883	15.094	8.64	5,910,044	.002714	19.195
92	111,859	19.313	28,676	864		.599	540	.889	12.638	8.69	5,402,431	.003200	17.678
93	113,072	19.313	20,942	851		.599	537	.887	10.124	8.53	5,563,433	.002838	18.771
94	113,262	19.313	22,434	838	954-	.599	536	.889	11.651	8.43	5,638,843	.002512	19.952
95	112,220	19.313	24,144	825	1,013-	.599	537	.891	12.279	8.44	5,580,491	.002636	19.477
96	112,342	19.313	31,233	809	3,712-	.599	538	.893	12.317	8.40	5,613,080	.003089	17.992
97	112,574	19.313	21,917	792	1,253+	.599	538	.897	11.291	8.34	5,665,177	.002854	18.718
98	109,244*	19.313	26,732	965		.608	530	.857	15.631	8.69	5,355,419	.002596	19.627
99	114,782*	19.313	32,600	994		.608	530	.859	15.094	8.79	5,562,912	.002982	18.312
100	111,168	19.313	25,117	989		.604	529	.861	12.638	8.71	5,401,379	.002930	18.474
101	113,447	19.313	18,960	977		.604	531	.864	10.124	8.75	5,486,952	.002639	19.466
102	111,956	19.313	21,448	966		.604	531	.865	11.651	8.71	5,439,603	.002634	19.485
103	111,834	19.313	23,881	955	1,435-	.604	531	.866	12.279	8.69	5,446,345	.002628	19.507
104	113,457	19.313	29,312	939	5,267-	.604	532	.870	12.317	8.63	5,563,809	.002750	19.069
105	112,268	19.312	19,499	928	2,250+	.595	531	.875	11.291	8.58	5,455,220	.002795	18.915
106	228,820	23.188	31,723	912	3,644-	.595	530	.876	9.918	8.54	9,303,388	.002469	20.125
107	227,273	23.188	29,795	892	3,326-	.595	530	.880	9.524	8.42	9,372,567	.002441	20.240
108	241,070	23.188	35,020	871	1,781-	.595	530	.882	10.492	8.37	10,000,979	.002459	20.166
109	233,605	23.188	43,310	846	7,287+	.595	529	.884	13.998	8.28	9,796,250	.002995	18.272
110	237,469	23.188	58,351	822	9,951-	.596	528	.885	14.846	8.18	10,097,005	.002609	19.578
111	235,325	23.188	32,568	794	6,960+	.596	529	.891	9.037	8.15	10,042,915	.003518	16.860
112	44,140	13.375	5,006	736		.604	528	.893	1.983	8.04	3,354,752	.004000	15.811

Test No.	Mcfd @ 14.73#	Internal Dia. Inch.	$P_1^2 - P_2^2$	P Avg.	Elevation Factor	Grav.	Temp. Abs.	Dev.	Length Pipe Mi.	Gas Visc.	Reynolds Number	Resist. Coef.	Trans. Fact.
113	68,543	15.375	23,992	736		.580	524	.900	10.123	8.00	4,373,659	.003000	18.257
114	28,431	12.125	21,801	727		.560	515	.903	15.995	7.75	2,292,753	.003667	16.514
115	102,926	19.313	25,391	725		.581	524	.901	15.180	7.90	5,303,861	.002909	18.541
116	103,968*	19.313	21,912	707		.579	524	.904	14.324	7.86	5,366,206	.002619	19.540
117	103,080*	19.313	25,255	686		.579	525	.907	16.700	7.83	5,340,828	.002592	19.642
118	131,625	19.313	31,303	979		.581	537	.882	12.882	9.05	5,920,872	.002590	19.649
119	128,799	19.313	38,323	960		.581	524	.872	16.333	8.48	6,183,245	.002681	19.313
120	131,623	19.313	24,602	943		.581	521	.872	9.048	8.37	6,401,670	.003024	18.185
121	132,649	19.313	26,284	931	1,718-	.584	521	.870	9.910	8.31	6,531,809	.002696	19.259
122	130,981	19.313	28,238	916	3,739-	.584	520	.870	9.682	8.25	6,496,579	.002795	18.915
123	132,346	19.313	42,073	897	584+	.584	520	.872	12.764	8.18	6,620,395	.003644	16.566
124	103,440*	19.313	21,868	975		.615	521	.848	15.631	8.45	5,275,139	.002444	20.228
125	102,984*	19.313	25,023	955		.615	519	.848	15.094	8.35	5,314,709	.002930	18.474
126	90,984*	13.375	66,516	1,116	1,545-	.615	543	.856	7.826	10.10	5,605,078	.002737	19.114
127	108,542	13.375	61,245	1,087	4,795-	.615	524	.836	4.856	8.85	7,631,241	.003000	18.257
128	108,542	13.375	119,935	1,046	8,354+	.615	520	.837	12.080	8.60	7,852,821	.002711	19.206
129	108,542	13.375	135,219	982	1,467+	.615	517	.851	11.584	8.40	8,040,142	.002973	18.340
130	108,542	13.375	154,849	905	1,201+	.615	517	.854	13.928	8.20	8,235,868	.002778	18.973
131	108,542	13.375	179,131	807	4,297-	.615	516	.867	15.418	8.00	8,442,150	.002800	18.898
132	138,963	13.375	227,469	1,038	1,710+	.606	544	.868	12.761	9.60	8,875,000	.002592	19.642
133	138,963	13.375	208,976	927	264+	.606	519	.857	12.494	8.15	10,453,628	.002585	19.668
134	138,963	13.375	251,006	788	3,443+	.606	514	.870	14.344	7.81	10,908,960	.002720	19.174
135	141,429	13.375	237,387	764	1,471+	.607	514	.878	12.343	7.78	11,163,656	.002824	18.818

Test No.	Mcfd @ 14.73#	Internal Dia. Inch	$P_1^2 - P_2^2$	P Avg.	Elevation Factor	Grav.	Temp. Abs.	Dev.	Length Pipe Mi.	Gas Visc.	Reynolds Number	Resist. Coef.	Trans. Fact.
136	141,429	13.375	179,658	613	19+	.607	512	.901	8.322	7.53	11,535,001	.003064	18.066
137	[illegible]429	13.375	246,103	770	1,494+	.607	514	.878	12.037	7.78	11,163,656	.003015	18.212
138	141,429	13.375	193,822	611	18+	.607	513	.901	8.322	7.53	11,535,001	.003298	17.413
139	102,460	15.438	46,758	831	5,153+	.572	541	.905	10.223	8.57	5,994,482	.002833	18.788
140	102,460	15.438	52,494	801	694-	.572	529	.900	11.031	8.15	6,303,211	.002656	19.404
141	102,460	15.438	68,129	763	1,385-	.572	529	.905	14.714	8.08	6,357,784	.002619	19.540
142	102,460	15.438	31,783	730	2,506+	.572	528	.907	7.356	8.03	6,397,274	.002667	19.364
143	102,460	15.438	70,68[illegible]	694	429+	.572	528	.912	15.538	7.95	6,461,908	.002600	19.611
144	102,460	15.438	51,067	648	484+	.572	528	.917	11.082	7.86	6,535,932	.002656	19.404
145	102,460	15.438	28,719	615	936+	.572	527	.920	6.683	7.81	6,577,673	.002579	19.691
146	104,000	17.375	28,538	747	597-	.570	529	.906	11.031	8.06	5,728,221	.002515	19.940
147	104,000	17.375	38,568	724	1,238-	.570	529	.908	14.714	8.03	5,749,570	.002523	19.909
148	104,000	17.375	16,496	705	2,319+	.570	529	.910	7.356	7.99	5,778,146	.002545	19.822
149	104,000	17.375	38,868	685	416+	.570	529	.913	15.523	7.94	5,814,584	.002543	19.830
150	104,000	15.438	31,761	615	936+	.570	527	.920	6.683	7.80	6,661,518	.002700	19.245
151	117,883	17.375	57,444	1,201	11,805-	.565	541	.872	13.906	10.63	4,879,697	.002647	19.437
152	117,883	17.375	29,228	1,184	9,694+	.565	535	.866	11.911	10.08	5,146,055	.002674	19.338
153	117,883	17.375	46,246	1,164	10,580-	.565	533	.865	10.743	9.87	5,255,583	.002718	19.181
154	117,883	17.375	21,778	1,150	365+	.565	532	.867	6.883	9.65	5,375,321	.002640	19.462
155	83,764	15.375	57,913	1,201	11,836-	.565	540	.871	13.693	10.50	3,966,923	.002960	18.380
156	83,764	15.375	26,315	1,185	10,376+	.565	534	.865	11.957	10.07	4,136,278	.002682	19.309
157	83,764	15.375	67,614	1,165	10,849-	.565	533	.866	17.560	9.88	4,215,786	.002844	18.751
158	201,580	15.375	122,343	1,125	7,604-	.568	530	.863	7.045	9.50	10,607,832	.002500	20.000

Test No.	Mcfd @ 14.73#	Internal Dia. Inch.	$P_1^2 - P_2^2$	P Avg.	Elevation Factor	Grav.	Temp. Abs.	Dev.	Length Pipe Mi.	Gas Visc.	Reynolds Number	Resist. Coef.	Trans. Fact.
159	201,580	15.375	201,783	1,050	864-	.568	530	.872	11.806	9.10	11,074,119	.002563	19.752
160	173,614	15.375	127,300	970	387+	.568	529	.879	9.427	8.66	10,022,005	.002747	19.080
161	173,614	15.375	58,672	921	6,980+	.568	529	.886	4.742	8.54	10,163,214	.002789	18.966
162	173,614	15.375	60,324	885	5,252-	.568	529	.889	3.928	8.34	10,406,535	.002781	18.963
163	168,142	15.375	119,769	832	6,154-	.568	527	.895	9.406	8.19	10,263,421	.002577	19.699
164	41,381	12.250	29,417	600	1,823-	.600	544	.918	10.007	8.09	3,390,313	.002800	18.898
165	41,381	12.250	28,096	576	1,394+	.600	534	.922	11.693	7.86	3,489,251	.002500	20.000
166	41,165	12.250	25,088	552	681+	.600	534	.925	10.144	7.85	3,475,666	.002600	19.611
167	36,508	10.250	52,443	514	1,140+	.600	534	.929	10.569	7.78	3,716,865	.002750	19.069
168	36,508	10.250	68,033	451	808+	.600	533	.937	13.160	7.67	3,770,287	.003000	18.257
169	36,508	10.250	71,625	365	1,369+	.600	533	.947	14.047	7.58	3,814,929	.002500	20.000
170	36,508	10.250	27,209	293	351+	.600	533	.959	5.360	7.52	3,845,615	.003000	18.257
171	120,668*	15.438	93,418	754	3,210+	.560	535	.912	15.452	8.19	7,231,968	.002607	19.585
172	125,740	17.375	53,688	733	3,039+	.560	535	.910	15.452	8.14	6,737,255	.002507	19.972
173	101,610*	17.375	34,535	697	2,732+	.560	535	.917	15.452	8.06	5,498,358	.002523	19.909
174	97,780*	15.438	62,347	712	2,905+	.570	535	.913	15.452	8.07	6,053,941	.002610	19.574
175	71,539*	15.438	33,184	674	2,594+	.570	535	.917	15.452	8.05	4,439,974	.002682	19.309
176	76,768*	17.375	19,445	668	2,539+	.570	535	.919	15.452	8.02	4,249,229	.002600	19.611
177	57,653*	17.375	10,938	650	2,411+	.570	535	.919	15.452	8.00	3,199,209	.002857	18.709
178	57,159*	15.438	21,330	657	2,462+	.570	535	.919	15.452	8.01	3,565,260	.002786	18.945
179	126,993*	15.438	104,619	764	3,367+	.570	535	.908	15.452	8.21	7,728,047	.002565	19.745
180	142,800	17.375	69,769	762	3,346+	.570	535	.908	15.452	8.21	7,721,346	.002494	20.024
181	134,544	19.250	48,372	448		.569	539	.947	13.634	7.78	6,916,939	.003333	17.321

Test No.	Mcfd @ 14.73#	Internal Dia. Inch	$P_1^2 - P_2^2$	P Avg.	Elevation Factor	Grav.	Temp. Abs.	Dev.	Length Pipe Mi.	Gas Visc.	Reynolds Number	Resist. Coef.	Trans. Fact.
182	134,544	19.250	39,360	396	130-	.569	539	.952	10.750	7.72	6,970,931	.003404	17.140
183	127,163	21.375	8,023	454		.571	538	.945	3.884	7.78	5,908,358	.003722	16.391
184	127,163	21.375	35,417	428		.571	538	.947	18.360	7.75	5,931,184	.003442	17.045
185	128,752	21.375	5,240	368		.579	536	.953	3.333	7.65	6,169,092	.002750	19.069
186	175,036	21.375	21,082	492		.583	556	.946	6.810	8.22	7,859,363	.002750	19.069
187	175,036	21.375	21,515	470		.583	545	.943	7.453	7.94	8,136,283	.002647	19.437
188	175,036	21.375	11,599	451		.583	543	.946	4.354	7.88	8,198,112	.002425	20.307
189	175,036	21.375	18,029	432		.583	541	.947	7.282	7.81	8,271,774	.002254	21.063
190	171,861	21.375	39,789	394	1,472-	.578	537	.951	14.507	7.69	8,177,952	.002520	19.920
191	171,861	21.375	22,421	353		.578	535	.955	8.996	7.63	8,242,136	.002397	20.425
192	171,861	21.375	23,655	319	479+	.578	536	.960	9.289	7.59	8,285,318	.002463	20.149
193	119,789	17.500	56,642	442		.569	539	.948	13.634	7.78	6,774,440	.003053	18.098
194	119,789	17.500	44,096	380	120-	.569	539	.955	10.750	7.70	6,844,824	.003000	18.257
195	165,917	21.375	9,515	452	121+	.571	538	.945	3.884	7.77	7,719,171	.002613	19.563
196	165,917	21.375	42,105	421		.571	534	.949	16.136	7.68	7,809,454	.002729	19.142
197	166,891	21.375	8,618	386		.579	536	.951	2.789	7.66	7,986,319	.003130	17.874
198	166,891	21.375	9,669	372		.579	535	.955	3.333	7.63	8,017,659	.003000	18.257
199	125,476	17.375	31,664	486		.583	556	.946	6.810	8.22	6,931,102	.002848	18.738
200	125,476	17.375	33,367	452		.583	544	.945	7.450	7.89	7,220,804	.002829	18.801
201	125,476	17.375	19,850	420		.583	539	.951	4.354	7.78	7,322,829	.002950	18.411
202	125,476	17.375	31,397	385	89-	.583	537	.951	7.280	7.68	7,418,315	.002735	19.121
203	121,856	21.375	31,201	347	1,115-	.578	537	.957	21.630	7.65	5,828,645	.002653	19.415
204	121,738	23.390	7,610	318		.578	535	.960	8.996	7.58	5,370,389	.002500	20.000

Test No.	Mcfd @ 14.73#	Internal Dia. Inch.	$P_1^2 - P_2^2$	P Avg.	Elevation Factor	Grav.	Temp. Abs.	Dev.	Length Pipe Mi.	Gas Visc.	Reynolds Number	Resist. Coef.	Trans. Fact.
205	121,737	23.500	8,044	306	441+	.578	536	.960	9.396	7.60	5,331,299	.002780	18.966
206	199,854	23.500	20,748	497		.571	540	.941	8.176	7.88	8,339,021	.002937	18.452
207	199,854	23.500	33,767	469		.571	539	.945	13.634	7.81	8,413,534	.002867	18.676
208	199,854	23.500	26,822	434	101-	.571	538	.948	10.760	7.75	8,478,669	.002872	18.660
209	255,524	17.375	107,863	1,189	4,408+	.563	552	.884	7.298	10.62	10,550,130	.002542	19.834
210	255,524	17.375	141,968	1,135	1,898-	.563	551	.885	8.872	10.30	10,877,906	.002616	19.551
211	255,524	17.375	112,629	1,078	253-	.563	547	.887	7.741	9.90	11,317,423	.002413	20.357
212	255,476	17.375	119,002	1,022	8,774+	.563	543	.889	8.713	9.50	11,791,773	.002445	20.224
213	253,633	17.375	127,009	960	4,017-	.563	538	.891	8.318	9.12	12,194,308	.002535	19.861
214	253,633	17.375	115,294	895	1,357+	.563	532	.891	8.106	8.43	13,192,531	.002489	20.044
215	250,087	17.375	107,323	829	555-	.568	530	.897	7.477	8.19	13,508,152	.002516	19.936
216	250,087	17.375	147,867	746	2,615+	.568	527	.904	10.704	7.93	13,951,299	.002464	20.145
217	253,490	17.375	27,065	1,189	1,523+	.568	553	.880	1.942	10.83	10,354,307	.002500	20.000
218	253,490	17.375	111,250	1,177	14,727-	.568	544	.874	6.752	10.62	10,559,126	.002444	20.228
219	248,378	17.375	189,335	1,111	1,174+	.568	538	.874	13.724	9.93	11,065,206	.002500	20.000
220	248,498	17.375	100,154	1,044	6,069+	.568	533	.877	7.694	9.30	11,820,038	.002500	20.000
221	236,824	17.375	104,428	991	6,229+	.567	530	.881	7.896	8.78	11,911,242	.002803	18.888
222	236,722	17.375	108,480	935	2,937+	.567	529	.887	7.916	8.55	12,225,902	.002805	18.881
223	236,728	17.375	78,602	884	1,452+	.567	527	.891	5.593	8.32	12,564,541	.002867	18.676
224	236,598	17.375	193,808	800	3,252-	.567	526	.899	14.141	8.10	12,898,465	.002665	19.371
225	231,832	17.375	76,639	711		.567	524	.909	5.638	7.85	13,041,719	.002768	19.007
226	114,288*	19.313	30,297	1,012		.618	529	.850	15.631	8.84	5,598,137	.002684	19.302
227	116,016*	19.313	35,835	993		.618	528	.851	15.094	8.73	5,754,567	.003214	17.639

Test No.	Mcfd @ 14.73#	Internal Dia. Inch.	$P_1^2 - P_2^2$	P Avg.	Elevation Factor	Grav.	Temp. Abs.	Dev.	Length Pipe Mi.	Gas Visc.	Reynolds Number	Resist. Coef.	Trans. Fact.
228	118,193	19.313	27,937	973		.594	532	.871	12.638	8.70	5,654,327	.002878	18.640
229	118,193	19.313	32,940	961	222-	.594	531	.871	10.124	8.67	5,673,913	.004231	15.374
230	118,193	19.313	24,466	947	1,251-	.594	531	.872	11.651	8.63	5,700,126	.002600	19.611
231	118,193	19.313	26,431	933	1,338-	.594	531	.874	12.279	8.56	5,746,673	.002681	19.313
232	120,120*	19.313	29,512	923	4,914-	.588	531	.878	12.317	8.50	5,822,186	.002531	19.877
233	120,120*	19.313	22,479	908	1,678+	.588	530	.880	11.291	8.45	5,856,793	.002711	19.206

* All volumes are as measured by ammonia tracer except those marked *, in which cases metered flow was used.

APPENDIX B

Correction for Elevation Change

In areas in which there are material differences in elevation it is necessary to adjust for these differences. Such adjustments have been made in the analysis of test data given in Appendix A. This correction has been made by the application of an adjusting factor, shown here as an adjustment to the differences of the squares of the absolute pressure in Equation 12.

$$Q = K_1 \left[\frac{(P_1^2 - P_2^2) - 0.0375 \dfrac{G(H_2 - H_1)\ Pavg}{Zavg\ .\ Tavg}^2}{L} \right]^{1/2} x\sqrt{1/Z} \qquad (14)$$

where the elevation factor term is

$$\frac{0.0375\ G\ (H_2 - H_1)\ Pavg^2}{Zavg\ .\ Tavg}$$

G = gravity of flowing gas (air = 1.000)
Z = average supercompressibility factor
T = average flowing temperature °F absolute
P = average flowing pressure psia
H_1 = elevation at inlet of pipe section (feet)
H_2 = elevation at outlet of pipe section (feet)

Calculation for Reynolds' Number

$$R = 135.321 \times 10^{-4} \times \frac{QG}{Vd} \qquad (15)$$

where

Q = Mcfd at 60 °F and 14.73 psia
G = gravity of gas (air = 1.000)
*V = viscosity of gas (lb. per sec. ft.)
d = internal diameter of pipe inches

Application of Deviation Factor

Beginning with the proposed formula (Equation 12)

$$Q = K_1 \left[\frac{(P_1^2 - P_2^2)}{L} \right]^{1/2} x \sqrt{1/Z} \qquad (12)$$

This formula expresses the deviation as a function of the mean effective pressure between the initial and terminal pressure of the span being

* V in Appendix A shown times 10^6.

considered. This formula may also be expressed

$$Q = K_1 \left[\frac{(P_1^2 - P_2^2) \quad 1/Z}{L}\right]^{1/2} \tag{16}$$

The literature suggests that

$$1/Z = (1 + JP_M) \tag{17}$$

where JP_M is the average deviation factor at the mean effective pressure and may be expressed as (1/Z - 1) for the pressure considered. P_M, the mean effective pressure, can be equated to P_1 and P_2 as follows.

$$P_M = 2/3 \left[\frac{P_1^3 - P_2^3}{P_1^2 - P_2^2}\right] \tag{18}$$

Substituting Equation 17 in Equation 16 gives

$$Q = K_1 \left[\frac{(P_1^2 - P_2^2) + JP_M (P_1^2 - P_2^2)}{L}\right]^{1/2} \tag{19}$$

Substituting Equation 18 for P_M in Equation 19 reduces to

$$Q = K_1 \left[\frac{(P_1^2 - P_2^2) + J\ 2/3 \left(\frac{P_1^3 - P_2^3}{P_1^2 - P_2^2}\right) (P_1^2 - P_2^2)}{L}\right]^{1/2} \tag{20}$$

Simplifying and grouping

$$Q = K_1 \left[\frac{(P_1^2 - P_2^2) + J\ 2/3 \quad (P_1^3 - P_2^3)}{L}\right]^{1/2} \tag{21}$$

$$Q = K_1 \left[\frac{(P_1^2 - P_2^2) + J\ 2/3\ P_1^3 - J\ 2/3\ P_2^3}{L}\right]^{1/2} \tag{22}$$

$$Q = K_1 \left[\frac{P_1^2\ (1 + J\ 2/3\ P_1) - P_2^2\ (1 + J\ 2/3\ P_2)}{L}\right]^{1/2} \tag{23}$$

Then, expressing

$$(1 + J\ 2/3\ P_1) = b_1^{\ 2} \tag{24}$$

Where b is a factor for the determination of the deviation of the gas at the

particular pressure being considered, the formula becomes:

$$Q = K_1 \left[\frac{(P_1^2 b_1^2) - (P_2^2 b_2^2)}{L} \right]^{1/2} \tag{25}$$

which formula is formula 13 above mentioned.

Thus, a table may be prepared making the appropriate adjustment for deviation of the gas considered, and relating the square of the absolute pressure adjusted for deviation to the corresponding gage pressure. This table will give factors which can be substituted in the proposed equation which will adjust for deviation.

APPENDIX C

Verification of Equation 15

"The Reynolds number, a dimensionless ratio commonly used to characterize the conditions of fluid flow in circular pipes, is given in the literature as:

$$R = \frac{Du\rho}{\mu} \text{ or } R = \frac{Du}{v} \tag{26}$$

where

R = Reynolds number;
D = Diameter of pipe;
u = average linear velocity;
ρ = density of fluid;
μ = dynamic viscosity of fluid;
v = kinematic viscosity of fluid (μ/ρ).

The above relationship has been expressed by Biddison[1] in natural-gas engineering terms as

$$R = 0.011459 \frac{QGP_b}{VdT_B}, \tag{27}$$

where

Q = rate of flow, cubic feet per hour at a pressure and temperature base of P_b p.s.i.a. and T_b, °F abs.;
G = specific gravity of gas (sp. gr. of air = 1.0);
V = viscosity of gas, pounds per second foot;
d = internal diameter of pipe, inches

The constant in equation (26) adjusts for the mixed units used in the equation."[2]

1. Biddison, P. M. Gas Flow Computations. Proc. Am. Gas Assoc. Nat. Gas Dept. 1941, pp. 51-88.
2. Bureau of Mines Monograph Number 9, Page 32 (Quotation taken from)

Equation 15 in our paper is obtained by substituting in Equation 26 above the following

$$Q = \frac{1000}{24} Q_1$$

$$P_b = 14.73 \text{ Psia}$$

$$T_b = (60^o \text{ F} + 460) = 520^o \text{ abs.}$$

where

Q_1 = rate of flow, thousand cubic feet per day (Mcfd) at a pressure and temperature base of 14.73 psia and 60° F abs.

Equation 26 becomes

$$R = 0.011459 \times \frac{14.73}{520} \times \frac{1000}{24} \times \frac{Q_1 G}{Vd}$$

Then

$$R = 135.321 \times 10^{-4} \quad \frac{Q_1 G}{Vd} \qquad \text{(Equation 15)}$$

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PREDICTION OF SURGE PRESSURES IN OIL PIPELINES[a]

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INTRODUCTION

Pressure surges have long been the bane of the pipeliner's existence. In spite of the advances in technology and equipment, surge problems for the most part have remained unsolved and are generally regarded as a necessary evil. undoubtedly, this has been so because the various water hammer theories were not fundamental enough in their approach to permit widespread application to a variety of pumping situations. In addition, many commercial desurgers or pulsation dampeners while helpful in some cases are frequently detrimental in others.

The net result is that the pipeliner is still concerned with:

1. The inability to predict the points of extreme fluctuating pressure in a pumping system, thereby resulting in the overdesign of the entire system.
2. Breakdowns of the pumping system from elastic failure of the pipes and pumps due to extreme fluctuating pressures.
3. Loss of revenue because "surges" necessitate the establishment of a line limit below the normal rating of the equipment.

It is the purpose of this paper to outline a method of analysis based on a fundamental mathematical approach and to demonstrate the validity of this method through application to an actual field-scale pumping pipeline system. The future applications of this method should materially aid the designer in predicting points of extreme fluctuating pressure, resulting in better overall designs and appreciable monetary savings.

Reduced to its most basic concept, pipelining is transportation. Transportation is movement, and moving objects have inertia and resistance.

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a. Presented at a meeting of the Pipeline Division, ASCE, February 18, 1957, Jackson, Miss.

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Therefore, each investigator in this field has succeeded only to the extent to which he has applied fundamental engineering techniques.

Development of a Method of Analysis

Previous Investigations

The earliest noteworthy contributions to the water hammer or pressure surge theory were those of Allievi[1] and Joukowski.[2] Joukowski postulated the classic water hammer formula for the increase in pressure due to the extinguishing of a given velocity of flow. The work of Allievi represented the first fundamental mathematical approach to variable flow. Although the basic differential equations of variable flow were not explicitly solved for the steady-state case until nearly fifty years later, the importance of Allievi's work should not be overlooked.

Many of the water hammer and pressure surge studies worthy of note draw heavily on the work of Allievi and Joukowski. The most significant contributions in this field have been summarized by the American Society of Mechanical Engineers Symposium on Water Hammer[3] and by Wyatt[4] and will not be repeated here. It should be noted, however, that for the most part the methods of analysis outlined in these papers were of a semi-empirical nature and lean heavily on graphical techniques.

Results of Recent Research

Mr. Waller[5,6] proposed a solution to the fundamental equations of variable flow postulated by Allievi,[1] Rayleigh,[7] Lamb,[8] and others. The basic differential equations found in this analysis were of the same form as those found in the transmission of energy by heat, sound, and electricity. It was natural, therefore, that use should be made of the mathematical tool of wave analysis. In the process of solution of the differential equations, the assumption was made that the principle of superposition was valid for this case. The resulting solution consisted of a transient term and a steady-state term.

The transient term was found to be the water hammer solution and obviously damps out after a period of time. Analyzing the steady-state term independently, the following result was obtained for the fluctuating pressure (P) and the alternating volumetric flow rate or volume current (Q) at any point "x" distance from the receiving end:

Note: Letter symbols adopted for use in this paper are completely defined and arranged in alphabetical order for convenience of reference in the appendix.

$$P_x = P_r \cosh \gamma x + Q_r Z_c \sinh \gamma x \tag{1}$$

and

$$Q_x = Q_r \cosh \gamma x + \frac{P_r}{Z_c} \sinh \gamma x \tag{2}$$

where Z_c and γ are constants defined as:

$$Z_c = \frac{\rho a}{A} \tag{3}$$

$$\gamma = \alpha + j\beta \tag{4}$$

where α is the attenuation coefficient and β is the phase constant. The reduction in magnitude of the wave as it travels down the pipeline may be determined from the attenuation coefficient:

$$\alpha = \frac{\bar{\beta} n}{2 \ell \bar{U} a} \tag{5}$$

Beta determines the change in phase between the pressure and volume current as the pressure wave travels in the pipeline:

$$\beta = \frac{\omega}{a} \tag{6}$$

The reader is referred to the original work for the complete derivation of the above equations. The primary interest herein is the application of this method of analysis to a full-scale field installation.

Eqs. 1 and 2 will be recognized as being of the same form as the classical wave equations of acoustics and electrical engineering, and an electrical-hydraulic analogy becomes evident. Therefore:

$$P_x = Q_x Z_x \tag{7}$$

i.e., ohms law for the hydraulic circuit becomes the defining equation for the impedance (Z) to the alternating flow. Eqs. 1 through 7 from the basis of the analysis of a pumping system. The recognition of the electrical analogy permits the formulation of a method of hydraulic circuit analysis similar in many respects to electric circuit analysis.

The Pump Equation

Inspection of Eq. 7 reveals that, once any two of the three quantities are known at a particular point in the pipeline system, the third may be calculated. Then, by application of Eqs. 1 and 2, the fluctuating pressure and alternating volume current may be calculated at other points in the system. Pressure is a measurable quantity. The volume current and impedance are not.

However, the use of a reciprocating pump on a pipeline system gives rise to an alternating periodic volume current or volumetric flow rate. Since the flow is periodic, it may be analyzed by Fourier Series methods and expressed in terms of a trigonometric series involving harmonics of a basic frequency. The response of the system to the various harmonics may then be determined; and, using superposition, the effects of all the harmonics may be found if desired.

When the connecting rod length, ℓ, is not relatively long compared to the crank arm, r, the piston motion is not simple harmonic, and the piston velocity for constant ω becomes:

$$v_p = \frac{ds}{dt} = r\omega \left[\sin\omega t + \frac{r}{2} \sin 2\omega t\right] \tag{8}$$

The variable flow rate is then:

$$Q = A_c v_p \tag{9}$$

where A_c is the area of the cylinder. For a double-acting cylinder $A_c = \frac{\pi D^2}{4}$ for $0 < \omega t < 180^\circ$ and $A_c = \frac{\pi(D^2 - d^2)}{4}$ for $180^\circ < \omega t < 360^\circ$ where "D" is the liner diameter and "d" is the diameter of the piston rod. Figure No. 1 is a plot of the discharge from one double-acting cylinder.

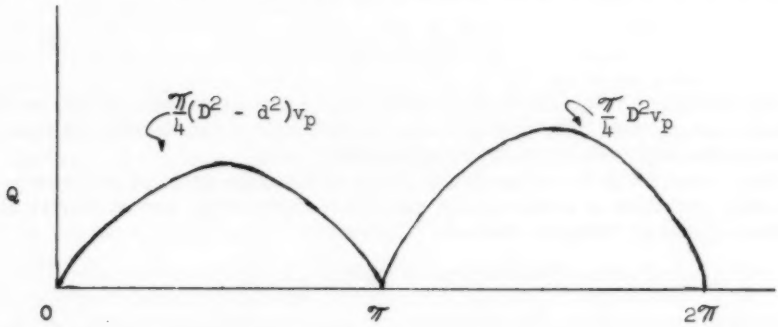

Fig. 1. Discharge from One Double-Acting Cylinder.

The Fourier Series solution for this case, referred to head end dead center, is:

$$\begin{aligned} Q = \frac{\pi r \omega}{4} \Bigg\{ & \frac{2D^2 - d^2}{\pi} - d^2 \left[\frac{2r}{3\pi l} \cos\omega t + \frac{1}{2} \sin\omega t\right] \\ & + \frac{(2D^2 - d^2)}{\pi} \left[\frac{\pi r}{4l} \sin 2\omega t - \frac{2}{3} \cos 2\omega t\right] \\ & + \frac{rd^2}{2\pi l} \sum_{m=3,5,7,9\ldots\ldots} \frac{4}{m^2 - 4} \cos m\omega t \\ & - \frac{(2D^2 - d^2)}{\pi} \sum_{m=4,6,8\ldots\ldots} \frac{2}{m^2 - 1} \cos m\omega t \Bigg\} \end{aligned} \tag{10}$$

The effect of referring Eq. 10 to crank end dead center is to reverse the algebraic sign of the odd harmonics.

For two double-acting cylinders, i.e., a double-acting duplex pump such as

the Gaso pumps at the Test Station with one cylinder ninety degrees ahead of the other, as shown in Figure No. 2, and with the first cylinder referred to

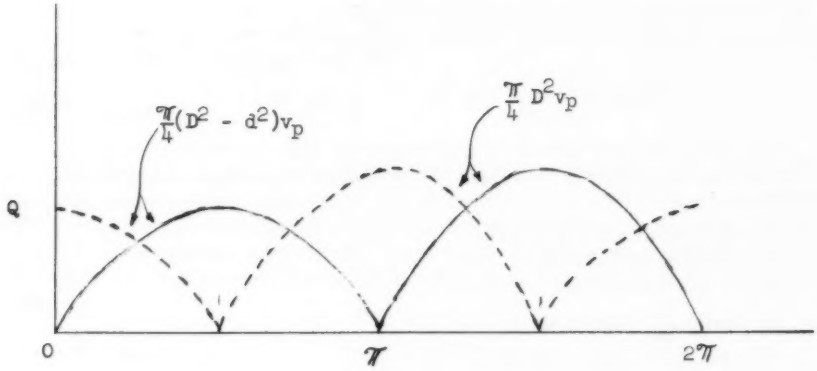

Fig. 2. Discharge from Double-Acting Duplex Pump.

head end dead center, the Fourier Series analysis gives:

$$Q = \frac{\pi r \omega}{4} \left\{ \frac{4D^2 - 2D^2}{\pi} - d^2 \left[\frac{r\sqrt{2}}{3\pi l} \operatorname{Sin}(\omega t + 135^\circ) + \frac{\sqrt{2}}{4} \operatorname{Sin}(\omega t + 45^\circ) \right] + \frac{rd^2}{2\pi l} \left[\sum_{\substack{m = 3, 7, 11 \ldots\ldots \\ m^2 - 4}} \frac{4\sqrt{2}}{} \operatorname{Cos}(m\omega t + 45^\circ) + \sum_{\substack{m = 5, 9, 13 \ldots\ldots \\ m^2 - 4}} \frac{4\sqrt{2}}{} \operatorname{Cos}(m\omega t + 135^\circ) \right] - \frac{(2D^2 - d^2)}{\pi} \cdot \left[\sum_{\substack{m = 4, 8, 12 \ldots\ldots \\ m^2 - 1}} \frac{4}{} \operatorname{Cos} m\omega t \right] \right\} \tag{11}$$

Similar equations may be developed for any multicylinder double-acting pump referenced to head end or crank end dead center by superimposing the effects of the required number of double-acting single cylinders (Eq. 10) properly phased.

It should be noted that the first term in Eqs. 10 and 11 is the mean flow ($\bar{q}$) component, while all the other terms are the variable flow components. The magnitude of all odd harmonics above the third and all even harmonics above the eighth is normally small in comparison to the predominant fourth harmonic. Neglecting these terms, Eq. 11 may be reduced to:

$$Q = \frac{\pi r \omega}{4} \left\{ \frac{4D^2 - 2d^2}{\pi} - d^2 \left[\frac{r\sqrt{2}}{3\pi \ell} \operatorname{Sin}(\omega t + 135^\circ) + \frac{\sqrt{2}}{4} \operatorname{Sin}(\omega t + 45^\circ) \right] + \frac{rd^2}{2\pi \ell} \frac{4\sqrt{2}}{5} \operatorname{Cos}(3\omega t + 45^\circ) - \frac{(2D^2 - d^2)}{\pi} \left[\frac{4}{15} \operatorname{Cos} 4\omega t + \frac{4}{63} \operatorname{Cos} 8\omega t \right] \right\} \tag{11a}$$

The following quantities are applicable to the Gaso pumps at the Test Station:

D = 5-1/2 inches = 0.458 ft.
d = 2 inches = 0.167 ft.
r = 6 inches = 0.5 ft.
$\omega = (\text{rpm})\frac{2\pi}{60} = 0.105\ (\text{rpm})$
ℓ = 33-1/2 inches = 2.742 ft.

Eq. 11a then reduces to:

$$\begin{aligned} Q = 0.0103\ (\text{rpm}) &+ 0.000816\ (\text{rpm})\ \operatorname{Sin}(\omega t - 128.9^\circ) \\ &+ 0.000221\ (\text{rpm})\ \operatorname{Sin}(3\omega t + 135^\circ) \\ &+ 0.00137\ (\text{rpm})\ \operatorname{Sin}(4\omega t - 90^\circ) \\ &+ 0.000325\ (\text{rpm})\ \operatorname{Sin}(8\omega t - 90^\circ) \end{aligned} \tag{11b}$$

Eq. 11b is used throughout the analyses in the following chapters as the value of the variable volume current, Q_g, generated by the pump.

For a pump with no return rod in the cylinder (i.e., d = 0), Eq. 10 for one double-acting cylinder becomes:

$$Q = \frac{\pi r \omega}{4} \left\{ \frac{2D^2}{\pi} + \frac{2D^2}{\pi} \left[\frac{\pi r}{4\ell} \operatorname{Sin} 2\omega t - \frac{2}{3} \operatorname{Cos} 2\omega t \right] - \frac{2D^2}{\pi} \sum_{m=4,6,8\ldots\ldots} \frac{2}{m^2 - 1} \operatorname{Cos} m\omega t \right\} \tag{12}$$

and Eq. 11, for two double-acting cylinders 90° apart, reduces to:

$$Q = \frac{\pi r \omega}{4} \left\{ \frac{4D^2}{\pi} - \frac{2D^2}{\pi} \sum_{m=4,8,12\ldots\ldots} \frac{4}{m^2 - 1} \operatorname{Cos} m\omega t \right\} \tag{13}$$

Equations may be developed for any multicylinder double-acting pump by superimposing the effects of the required number of double-acting cylinders (Eq. 12) properly phased.

Field Tests

The Test Station was equipped with four double-acting duplex, Gaso pumps with V-belt drive. Crude oil is pumped from tankage through approximately forty-five miles of eight-inch line into tankage. There are several diameter changes and short stub lines involved.

Instrumentation

Fluctuating pressures were measured and recorded with the aid of Consolidated Engineering Corporation and Baldwin pressure cells. The output of the cells was fed into a Midwestern recording oscillograph. The measured values of pressure recorded in Tables II through VII were obtained from the analysis of the oscillograph traces. Brown recording pressure gages were used to measure the mean discharge pressures at the sending and receiving ends.

Crude Oil Data

Samples of crude oil were taken before, during, and after the test runs. Analysis of the samples gave the following data:

Density 40.7° API at 54° F. or 1.59 slugs per cubic foot.
Viscosity 51.5 SUS or 13.45×10^5 lb. sec. per square foot.

Velocity of Propagation

The velocity of propagation of the pressure waves in the system was determined by measuring the time required for a down surge to travel from the sending end pumps to the receiving end. An average time of travel of 59.3 seconds was used, giving a velocity of propagation of 4029 feet per second.

Determination of the Attenuation Coefficient

The attenuation coefficient may be readily determined from Eq. 5. Pertinent data are tabulated in Table I. The mean pressure drop was plotted against the mean velocity and a value of 1.75 determined for "n." The resulting values of the attenuation coefficient were small enough to be ignored in the analysis of the Test Station piping.

Analysis

With the distance, x, measured from the receiving end, the pressure and volume current for any end termination, Z_r, are given by Eqs. 1 and 2.

With αx small (i.e., $\cosh \alpha x \doteq 1$ and $\sinh \alpha x \doteq 0$), Eqs. 1 and 2 reduce to:

$$P_x = P_r \cos \beta x + j Q_r Z_c \sin \beta x \tag{14}$$

and

$$Q_x = Z_r \cos \beta x + j \frac{P_r}{Z_c} \sin \beta x \tag{15}$$

At the receiving end:

$$P_r = Q_r Z_r \tag{16}$$

TABLE I

DATA FOR DETERMINATION OF n AND α

Test No.	$\bar{q}$ c.f.s.	$\bar{U}$ f.p.s.	PS* psi	PR* psi	PS-PR psi	$\bar{p}$** psi	$\alpha \times 10^{-6}$ Numeric per ft.	N_R $\times 10^{-4}$
1	0.797	2.29	111.0	---	---	---	9.339	1.800
2	0.847	2.44	138.0	8.3	129.7	289.5	9.945	1.920
3	0.687	1.98	67.0	7.5	59.5	219.3	8.309	1.558
4	0.612	1.76	60.0	7.3	52.7	212.5	7.712	1.383
5	0.787	2.27	90.0	7.5	82.5	242.3	9.241	1.785
6	0.870	2.50	148.0	8.5	139.5	299.3	10.036	1.965

* PS = mean pressure as shown by recorder at the sending end.

PR = mean pressure as shown by recorder at the receiving end.

** $\bar{p}$ = PS less PR plus effects of elevation change between stations.

Letting $x = \ell$, the sending end impedance becomes:

$$Z_s = Z_c \frac{Z_r + jZ_c \mathrm{Tan}\beta\ell}{Z_c + jZ_r \mathrm{Tan}\beta\ell} = \frac{P_s}{Q_s} \tag{17}$$

With the distance, x, measured from the sending end, the pressure and volume current for any end termination, Z_s, are given as:

$$P_x = P_s \mathrm{Cosh}\,\gamma x - Q_s Z_c \mathrm{Sinh}\,\gamma x \tag{18}$$

and

$$Q_x = Q_s \mathrm{Cosh}\,\gamma x - \frac{P_s}{Z_c} \mathrm{Sinh}\,\gamma x \tag{19}$$

Again, when αx is small (i.e., $\mathrm{Cosh}\alpha x \doteq 1$ and $\mathrm{Sinh}\alpha x \doteq 0$), Eqs. 18 and 19 reduce to:

$$P_x = P_s \mathrm{Cos}\beta x - jQ_s Z_c \mathrm{Sin}\beta x \tag{20}$$

and

$$Q_x = Q_s \mathrm{Cos}\beta x - j\frac{P_s}{Z_c} \mathrm{Sin}\beta x \tag{21}$$

Letting $x = \ell$ and

$$P_s = Z_s Q_s \tag{22}$$

the receiving end impedance, Z_r, becomes:

$$Z_r = Z_c \frac{Z_s - jZ_c \mathrm{Tan}\beta\ell}{Z_c - jZ_s \mathrm{Tan}\beta\ell} \tag{23}$$

The field test pipeline system was basically composed of pumps, pipeline, and receiving end tank, as shown in Figure No. 3a.

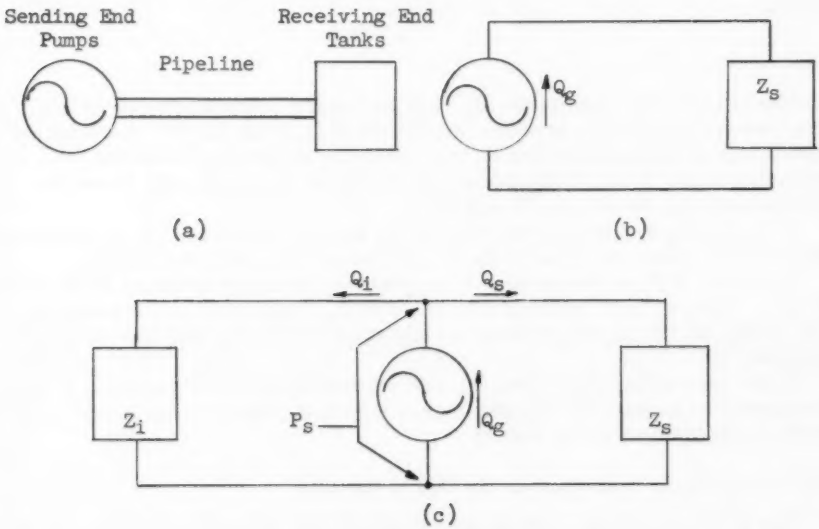

Fig. 3. Analysis Sketches.

Electrically, this system may be diagramed as shown in Figure No. 3b. Analysis of the system based on this arrangement gave pressure magnitudes in good agreement with measured values. However, the resulting phase angles were generally not in good agreement with measured values.

The concept of treating the pump as a constant current generator (electrically speaking) was then introduced. That is, the pump was thought of as having an internal impedance, Z_i, in parallel and generating a volume current, Q_g, which divides—part, Q_i, circulating through the internal impedance and the remainder, Q_s, flowing out into the pipeline system proper. See Figure No. 3c.

Then:

$$P_s = Z_s Q_s = Z_i Q_i = Z_p Q_g \tag{24}$$

and

$$Q_g = Q_i + Q_s \tag{25}$$

From Eqs. 24 and 25:

$$Z_p = \frac{Z_i Q_i}{Q_g} = \frac{Z_i Q_i}{Q_i + Q_s} = \frac{Z_i Z_s}{Z_i + Z_s} \tag{26}$$

or

$$Z_i = \frac{Z_p Z_s}{Z_s - Z_p} \tag{27}$$

The sending end impedance, Z_s, may be readily calculated from Eq. 17. The measurement of P_s permits calculation of Q_s from Eq. 24. Knowing Q_s and Q_g (from Pump Equation), Q_i may be easily determined from Eq. 25. With the values of Q_i and Q_g known, the values of Z_i and Z_p may be established from Eq. 24 or Eqs. 26 and 27.

If P_s and Z_s are known, the value of Q_s may be calculated. It is not necessary to determine the value of Z_i in order to proceed with the analysis.

However, if P_s is now known, it becomes necessary to evaluate Z_i by some means. This, in turn, permits evaluation of Z_p (impedance at the pump) by use of Eq. 26. Then, the sending end pressure value, P_s, may be calculated from Eq. 24.

In the case of the field tests, P_s was a known (measured) quantity. It was then possible to calculate Q_s and, hence, values of pressure and volume current at other points in the system.

One-Pump Tests with Pump at the End of the Line

The physical arrangement of the equipment was as shown in Figure No. 4. Points P2, P3, etc., denote locations of pressure pickups.

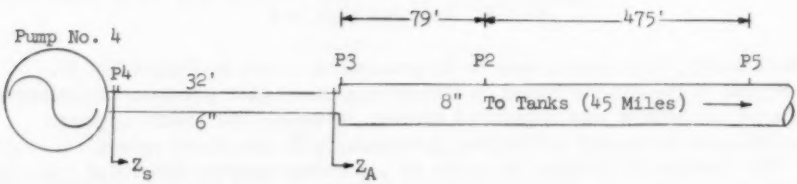

Figure No. 4 - Pump at End of Line

Three tests (Nos. 1, 2, and 3) at 66.7, 77.4, and 82.2 r.p.m. were run using this setup. The circuit may be visualized electrically as shown in Figure No. 3.

Applying the equations and techniques outlined in the previous section, the pressures at the various pickup points were calculated. Measured and calculated values are compared in Tables II, III, and IV.

TABLE II: Test No. 1*

COMPARISON OF MEASURED AND CALCULATED PRESSURE VALUES

	P4 (at Pump) Measured		Q_s (at Pump) Calculated		P3 (x = 32 feet) Calculated		P3 (x = 32 feet) Measured	
Harmonic	psi	Ø deg.	c.f.s.	Ø deg.	psi	Ø deg.	psi	Ø deg.
1	12.62	- 90.0	0.0980	- 85.8	12.77	- 96.2	12.50	-100.5
3	6.46	+145.8	0.0481	+133.5	6.23	+127.1	5.11	+122.2
4	16.46	- 90.0	0.1187	-105.4	16.72	-113.6	15.33	-118.6
8	5.18	-104.1	0.0304	-130.5	4.26	-148.2	5.17	-150.4
Prms.	15.79				15.80		14.90	

	P2 (x = 111 feet) Calculated		P2 (x = 111 feet) Measured		P5 (x = 554 feet) Calculated		P5 (x = 554 feet) Measured	
Harmonic	psi	Ø deg.	psi	Ø deg.	psi	Ø deg.	psi	Ø deg.
1	12.76	-105.2	11.44	-106.4	12.60	-159.7	9.89	-161.1
3	6.23	+ 99.8	4.97	+ 90.0	6.23	-115.6	4.95	- 92.6
4	16.25	-148.2	15.61	-156.8	15.57	- 8.4	12.27	- 27.6
8	4.40	+138.0	4.80	+143.8	4.25	+ 2.6	4.12	+ 33.7
Prms.	15.58		14.54		15.13		12.04	

* Pump No. 4 @ 77.4 r.p.m.

TABLE III: Test No. 2*

COMPARISON OF MEASURED AND CALCULATED PRESSURE VALUES

	P4 (at Pump) Measured		Q_s (at Pump) Calculated		P3 (x = 32 feet) Calculated		P3 (x = 32 feet) Measured	
Harmonic	psi	Ø deg.	c.f.s.	Ø deg.	psi	Ø deg.	psi	Ø deg.
1	12.73	- 94.7	0.0988	- 99.1	12.67	-101.4	12.39	- 98.8
3	7.17	+140.4	0.0529	+127.5	6.70	+125.2	6.55	+117.2
4	15.86	- 84.3	0.1129	-101.0	14.82	-110.2	15.17	-118.6
8	5.15	-103.4	0.0293	-130.5	4.15	-153.4	5.72	-155.7
Prms.	15.68				14.87		15.15	

	P2 (x = 111 feet) Calculated		P2 (x = 111 feet) Measured		P5 (x = 554 feet) Calculated		P5 (x = 554 feet) Measured	
Harmonic	psi	Ø deg.	psi	Ø deg.	psi	Ø deg.	psi	Ø deg.
1	12.68	-111.1	12.66	-108.7	12.67	-169.2	10.97	-169.9
3	6.70	+ 96.2	6.82	+ 91.3	6.70	- 78.2	5.94	-102.8
4	14.82	-148.9	13.12	-151.6	14.67	- 21.4	14.46	- 42.9
8	4.15	+129.2	5.09	+128.6	4.15	+ 24.2	3.71	- 7.4
Prms.	14.87		14.23		14.79		13.76	

* Pump No. 4 @ 82.2 r.p.m.

TABLE IV: Test No. 3*

COMPARISON OF MEASURED AND CALCULATED PRESSURE VALUES

	P4 (at Pump) Measured		Q_s (at Pump) Calculated		P3 (x = 32 feet) Calculated		P3 (x = 32 feet) Measured	
Harmonic	psi	Ø deg.	c.f.s.	Ø deg.	psi	Ø deg.	psi	Ø deg.
1	12.22	-115.9	0.0950	-119.8	12.17	-121.4	13.05	-118.8
3	5.95	+153.5	0.0448	+142.8	5.80	+137.2	5.50	+143.6
4	13.13	- 83.4	0.0963	- 97.4	12.52	-104.7	13.55	-106.2
8	4.95	- 79.1	0.0310	-103.2	4.23	-118.6	4.50	-120.4
Prms.	13.82				13.35		14.22	

	P2 (x = 111 feet) Calculated		P2 (x = 111 feet) Measured		P5 (x = 554 feet) Calculated		P5 (x = 554 feet) Measured	
Harmonic	psi	Ø deg.	psi	Ø deg.	psi	Ø deg.	psi	Ø deg.
1	12.17	-129.2	13.76	-124.2	12.17	-176.4	12.17	-151.2
3	5.80	+113.7	5.84	+111.5	5.80	- 27.8	5.71	- 50.5
4	12.53	-136.1	15.32	-136.7	12.52	+ 35.3	12.40	+ 24.8
8	4.23	+178.6	4.39	+169.7	4.23	+161.4	4.00	+133.5
Prms.	13.35		15.45		13.35		13.24	

* Pump No. 4 @ 66.7 r.p.m.

One-Pump Tests with Pump Not at the End of the Line

The physical arrangement of the equipment was as shown in Figure No. 5.

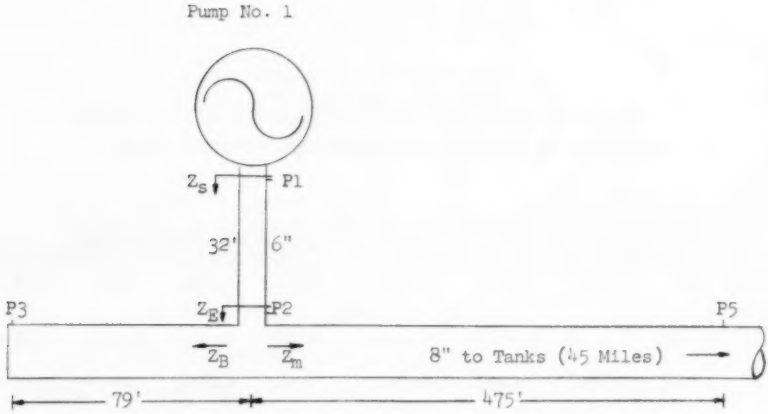

Figure No. 5 - Pump Not at End of Line

Three tests (Nos. 4, 5, and 6) at 59.4, 76.4, and 84.5 r.p.m. were run using this setup. The circuit may be visualized electrically as shown in Figure No. 3. However, Z_S in this case is the parallel combination of the long line and the short stub in series with the short six-inch line.

Again, applying the equations and techniques previously outlined, the pressures at the various pickup points were calculated. Measured and calculated pressure values are compared in Tables V, VI, and VII.

Analysis of Test Results

There were six one-pump tests. The oscillograph traces were analyzed for the first, third, fourth, and eighth harmonics at three different points in the pipeline.

In order for the theory to predict accurately the experimental pressure, the difference "Y" between the observed pressure and theoretical pressure, when analyzed statistically, must have reasonable values when compared to the precision of measurement. The differences $Y = P_{th} - P_{exp}$ are recorded in Table VIII.

When these 72 values are analyzed, it is found that:

1. "Y" is normally distributed with mean, $\overline{Y} = -0.15$ psi = 0.
2. The standard deviation $= \left[\frac{(Y_i = \overline{Y})^2}{72}\right]^{\frac{1}{2}} = 1.33$ psi.
3. Therefore, with 95 per cent confidence (i.e., 19-1 odds), the experimental pressure amplitudes are predicted by the theory within 2.66 psi (two standard deviations) of the theoretical pressure, without regard to the precision of measurement. It is noted that only one measurement differed by more than ± 4 psi and only eight by more than ± 2 psi. (See Table VIII.)

TABLE V: Test No. 4*

COMPARISON OF MEASURED AND CALCULATED PRESSURE VALUES

	P1 (at Pump) Measured		Q_s (at Pump) Calculated		P2 (x = 32 feet) Calculated		P2 (x = 32 feet) Measured	
Harmonic	psi	Ø deg.	c.f.s.	Ø deg.	psi	Ø deg.	psi	Ø deg.
1	8.70	- 90	0.0687	- 90	8.72	- 94.9	8.61	91.4
3	4.18	+117.5	0.0378	-126.6	5.28	+123.4	4.20	+ 99.6
4	8.54	-106.5	0.0920	- 95.6	9.94	-129.8	10.23	-130.6
8	1.40	- 56.3	0.0312	- 98.9	2.03	-160.9	1.79	-166.9
Prms.	9.16				10.17		9.99	

	P3 (x = 111 feet) Calculated		P3 (x = 111 feet) Measured		P5 (x = 507 feet) Calculated		P5 (x = 507 feet) Measured	
Harmonic	psi	Ø deg.	psi	Ø deg.	psi	Ø deg.	psi	Ø deg.
1	8.80	- 94.4	8.94	- 95.2	8.68	-136.4	7.89	-140.4
3	5.70	+123.4	4.62	+101.8	5.30	- 2.6	3.19	- 49.4
4	11.41	-129.8	12.17	-128.7	9.94	+ 62.2	8.86	+ 54.5
8	3.90	-160.4	3.72	-159.9	2.07	-184.8	1.86	-162.1
Prms.	11.30		11.47		10.16		8.78	

* Pump No. 1 @ 59.4 r.p.m.

TABLE VI: Test No. 5*

COMPARISON OF MEASURED AND CALCULATED PRESSURE VALUES

	P1 (at Pump) Measured		Q_s (at Pump) Calculated		P2 (x = 32 feet) Calculated		P2 (x = 32 feet) Measured	
Harmonic	psi	Ø deg.	c.f.s.	Ø deg.	psi	Ø deg.	psi	Ø deg.
1	11.92	-131.2	0.0962	-126.2	12.09	-137.6	12.48	-141.0
3	3.77	+136.2	0.0395	+146.6	4.32	+114.1	4.39	+ 74.8
4	7.21	-108.4	0.0969	-100.2	9.27	-143.7	12.83	-161.3
8	4.10	- 21.4	0.0472	-106.0	1.59	+176.3	1.14	+146.0
Prms.	10.60				11.26		13.05	

	P3 (x = 111 feet) Calculated		P3 (x = 111 feet) Measured		P5 (x = 507 feet) Calculated		P5 (x = 507 feet) Measured	
Harmonic	psi	Ø deg.	psi	Ø deg.	psi	Ø deg.	psi	Ø deg.
1	12.25	-137.6	13.72	-133.5	12.09	-191.6	12.74	-174.2
3	4.91	+114.1	5.28	+ 87.2	4.32	- 47.9	3.77	-101.6
4	11.73	-143.7	16.67	-148.6	9.27	+ 0.2	9.77	- 24.5
8	6.48	+176.3	6.28	+163.3	1.59	+104.1	5.09	+ 85.5
Prms.	13.30		16.33		11.25		11.97	

* Pump No. 1 @ 76.4 r.p.m.

TABLE VII: Test No. 6*

COMPARISON OF MEASURED AND CALCULATED PRESSURE VALUES

	P1 (at Pump) Measured		Q_s (at Pump) Calculated		P2 (x = 32 feet) Calculated		P2 (x = 32 feet) Measured	
Harmonic	psi	Ø deg.	c.f.s.	Ø deg.	psi	Ø deg.	psi	Ø deg.
1	7.87	-114.9	0.0639	-109.4	8.01	-122.0	8.21	-125.9
3	3.88	+115.7	0.0437	+126.3	4.62	+ 90.2	5.90	+ 97.2
4	7.59	-104.3	0.1159	- 99.8	10.59	-146.4	12.14	-142.7
8	7.82	- 42.6	0.0629	-131.6	0.83	+146.4	0.23	-135.0
Prms.	9.89				9.96		11.20	

	P3 (x = 111 feet) Calculated		P3 (x = 111 feet) Measured		P5 (x = 507 feet) Calculated		P5 (x = 507 feet) Measured	
Harmonic	psi	Ø deg.	psi	Ø deg.	psi	Ø deg.	psi	Ø deg.
1	8.15	-122.0	9.61	-131.7	8.01	-181.2	10.40	-178.1
3	5.40	+ 90.2	6.78	+ 94.8	4.61	- 89.5	4.69	- 99.9
4	14.18	-146.4	15.39	-146.2	10.59	- 25.4	9.94	- 20.4
8	7.38	+146.4	7.94	+152.9	0.83	+ 28.4	0.51	- 35.5
Prms.	13.25		14.80		9.96		10.71	

* Pump No. 1 @ 84.5 r.p.m.

TABLE VIII

DIFFERENCE BETWEEN MEASURED AND CALCULATED PRESSURE VALUES FOR EACH HARMONIC

Test	Pickup	Harmonic	Y Difference psi*	Test	Pickup	Harmonic	Y Difference psi*
1	3	1	+0.27	2	3	1	+0.29
		3	+1.12			3	+0.15
		4	+1.39			4	-0.35
		8	-0.91			8	-1.57
	2	1	+1.32		2	1	+0.02
		3	-1.26			3	-0.12
		4	+0.64			4	+1.70
		8	-0.40			8	-0.94
	5	1	+2.71		5	1	+1.70
		3	+1.28			3	+0.76
		4	+3.30			4	+0.21
		8	+0.13			8	+0.44
3	3	1	-0.88	4	2	1	+0.11
		3	+0.30			3	+1.08
		4	-1.03			4	-0.29
		8	-0.27			8	+0.24
	2	1	-1.59		3	1	-0.14
		3	-0.04			3	+1.08
		4	-2.79			4	-0.76
		8	-0.16			8	+0.18
	5	1	---		5	1	+0.79
		3	+0.09			3	+2.11
		4	+0.12			4	+1.08
		8	+0.23			8	+0.21
5	2	1	-0.39	6	2	1	+0.20
		3	-0.07			3	-1.28
		4	-3.56			4	-1.55
		8	+0.45			8	+0.60
	3	1	-1.47		3	1	-1.46
		3	-0.31			3	-1.38
		4	-4.94			4	-1.21
		8	+0.20			8	-0.56
	5	1	-0.65		5	1	-2.39
		3	+0.55			3	-0.08
		4	-0.50			4	+0.65
		8	-3.50			8	+0.32

* Plus indicates calculated value greatest.

TABLE IX

COMPARISON OF MEASURED AND CALCULATED
ROOT MEAN SQUARE PRESSURE VALUES

Test No.	Pickup No.	Calculated Prms psi	Measured Prms psi	Difference psi*
1	3	15.80	14.90	+0.90
1	2	15.58	14.54	+1.04
1	5	15.13	12.04	+3.09
2	3	14.87	15.15	-0.28
2	2	14.87	14.23	+0.64
2	5	14.79	13.76	+1.03
3	3	13.35	14.22	-0.87
3	2	13.35	15.45	-2.10
3	5	13.35	13.24	+0.11
4	2	10.17	9.99	+0.18
4	3	11.30	11.47	-0.17
4	5	10.16	8.78	+1.38
5	2	11.26	13.05	-1.79
5	3	13.30	16.33	-3.03
5	5	11.25	11.97	-0.72
6	2	9.96	11.20	-1.24
6	3	13.25	14.80	-1.55
6	5	9.96	10.71	-0.75

* Plus indicates calculated value greater.

When considering that the factors that comprise the error of measurement alone might be as much as ±4.32 psi, one can only conclude the fundamental principles and theory satisfactorily predict the measured pressures well within the limits of the instrumentation.

In comparing the root mean square (square root of one-half the sum of the squares) pressure values, it was found that none of the eighteen values exceeded the probable error of ±4.32 psi. (See Table IX.)

CONCLUSIONS AND RECOMMENDATIONS

The results of the work related herein adequately account for the pressure fluctuations in a reciprocating pump system. Specifically:

1. The validity of the basic underlying derivations has been established.
   a) The fundamental principles and equations formulated in the basic theoretical analysis have been verified.
   b) The principle of superposition is applicable to this situation.
   c) Simplification of the general mathematical expressions is valid for this case.

2. The applicability of the fundamental principles to a field-scale system has been demonstrated.
   a) The techniques of analysis were successfully applied to an actual field installation.
   b) It appears that the use of these results will be limited only by the ingenuity of the persons performing the analysis.
   c) The results of this work are sufficiently general enough in nature to permit the designer to apply them to a variety of piping configurations with a high degree of confidence.

The parameters based on the concepts of the hydraulic-electric analogy have been reasonably well established. Although work should be continued on the study of these basic parameters, the main effort should be directed toward further development of techniques of analysis for more complex pipeline systems.

The basic theme of this entire research activity is the hydraulic-electric analogy. This being the case, electronic computer techniques could be applied and would greatly facilitate the preparation of much of the material in chart and graph form. Such procedures could lessen considerably the computational effort required in the analysis of many pressure surge problems.

ACKNOWLEDGMENTS

The work reported herein was performed by the staff of the Division of Engineering Research, Oklahoma Institute of Technology, Oklahoma A&M College, Stillwater, Oklahoma. Future work plans include application of digital computer techniques to the analysis and the use of the method of analysis in analyzing corrective devices.

The study was performed under the administrative direction of Dr. Clark A. Dunn, Director, Division of Engineering Research. Dr. H. T. Fristoe and Professors R. L. Flanders and J. R. Norton rendered valuable assistance in related consulting capacities.

Many persons contributed their talents to achieve the results summarized herein; it is not possible to acknowledge specifically the contribution which each individual made. However, the writers wish to gratefully acknowledge the splendid cooperation of those who made this research possible.

APPENDIX - NOTATION

The symbols listed below are briefly defined, and the descriptive units refer to the gravitational (technical) system. "F" is a force usually in pounds, "L" is a length usually feet, and "T" is time usually seconds. The mass unit is the slug = 32.2 lb/g.

- a - The velocity of propagation of the pressure wave. LT^{-1}, or ft. per sec.
- A - Area. L^2, square ft.
- A - As a subscript refers to a particular point in pipeline system; see appropriate sketches.
- B - As a subscript refers to the stub or branch in the pipeline system.
- d - The diameter of the piston rod. L, ft.
- D - The diameter of the pipe for the pipeline analysis. L, ft.
- D - The diameter of the liner for the pump analysis. L, ft.
- E - As a subscript refers to equivalent parallel combination of branch and main line impedances.
- f - The frequency. T^{-1}, cycles per second.
- g - The gravitation constant. LT^{-2}, 32.2 ft. per sec^2.
- i - As a subscript refers to internal mechanism of pump.
- j^2 - 1, i.e., j is complex operator = $\sqrt{-1}$.
- ℓ - Length. L, ft.
- m - An integer.
- m - As a subscript refers to the main line.
- n - The exponent of $\overline{U}$ for mean flow.
- N_r - The Reynold's Number.
- p - As a subscript refers to the equivalent impedance at the pump (i.e., parallel combination of sending end and internal pump impedances).
- $\overline{p}$ - The mean pressure that causes a mean flow in a finite length of line. FL^{-2}, lb. per ft.^{2}.
- P_{sub} - The vectorial representation of the standing wave pressure amplitude. The subscript denotes the point in question. FL^{-2}, lb. per ft.^{2}.
- $\overline{q}$ - The mean rate of flow. L^3T^{-1}, cubic ft. per sec.
- Q_{sub} - The vectorial representation of the volume current or harmonic flow rate. L^3T^{-1}, cubic ft. per sec.
- Q_g - The volume current generated by the oscillating piston of the reciprocating pump. L^3T^{-1}, cubic ft. per sec.
- r - As a subscript refers to the receiving end of the pipeline.
- r - The radius of the crank arm or the eccentric of the crankshaft. L, ft. or 1/2 of stroke.
- s - As a subscript refers to the sending end of the pipeline. This is usually at the pump end.
- s - The stroke of the piston (2r). L, ft.

t	-	Time. T, sec.
$\bar{U}$	-	The mean velocity of the fluid in the pipeline. LT^{-1}, ft. per sec.
v_p	-	The velocity of the piston in a reciprocating pump. LT^{-1}, ft. per sec.
x	-	The distance from the receiving end or sending end. See appropriate sketches. L, ft.
x	-	Used as a second subscript to denote existence of several harmonics.
Z_c	-	The characteristic impedance of the pipeline. $FL^{-5}T$, lb. sec. per ft.^{5}.
Z_{sub}	-	The complex quotient of the hydromotive pressure (P) and hydraulic volume current (Q) at any point in the system denoted by the subscript. $FL^{-5}T$, lb. sec. per ft.^{5}.
α	-	The attenuation constant. L^{-1}, numeric per ft.
β	-	The phase constant. L^{-1}, radians per ft. or degrees per ft.
γ	-	The propagation constant $\alpha + \beta$. L^{-1}.
ρ	-	The density of the fluid. $FL^{-1}T^2$ per L^3 or $FL^{-4}T^2$, slugs per ft.^{3}.
ϕ	-	The phase angle of the polar form of the vectors (degrees).
ω	-	The circular speed of the rotating vectors; also the angular velocity of the crankshaft. L^{-1}, radians per second.
$\doteq$	-	This sign as used herein means approximately equals.

NOTE: All capital P's, Q's, and Z's in equations in this report refer to vector quantities. When a single subscript is used, it refers to a particular position. When two subscripts are used, the first refers to position, the second to the particular harmonic in question.

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DIVISION ACTIVITIES
PIPELINE DIVISION

Proceedings of the American Society of Civil Engineers

NEWS

March, 1957

The PIPELINE DIVISION was inaugurated at the annual convention in Pittsburgh last October. At these first meetings each chairman of the administrative and technical committees was assigned the responsibility for organizing his committee to work with Local Sections, the Junior Members and Student Chapters. Also, to correlate his task groups with work done by other divisions of the Society, the API, AGA, and ASME in particular, other societies and organizations, to avoid any duplication of effort. At the Jackson, Mississippi, convention in February, task groups with schedules for this year's accomplishments were reported by all chairmen.

Local Sections have been contacted requesting their nominations of one or more co-ordinators for our Pipeline Division activities. Also, for a list of members who desire affiliation with our division, since this information is not available at Headquarters. This will add to our mailing list for our JOURNAL and division notices. If you know of anyone who should affiliate with us, please urge them to advise Headquarters direct.

Volunteer workers are still needed for all of the administrative and technical committees. If you are interested, please communicate with the committee chairmen listed below, or any member of the Executive Committee:

Publications - H. A. Smallwood, Assoc. Professor of Civil Engineering, Madison, Wisconsin.

Session Programs - R. E. Kling, Chief Engineer, Site Oil Company, 3010 Locust Street, St. Louis 3, Missouri.

Membership - L. M. Odom, Pyburn & Odom, Consultants, P. O. Box 267, Baton Rouge, Louisiana.

Public Relations - D. M. Taylor, Gulf Coast Editor, "The Petroleum Engineer," Rice Professional Bldg., 2370 Rice Blvd., Houston, Texas.

Cooperation With Local Sections - R. R. Crawford, Senior Engineer, Bechtel Corp., 209 H Street, San Rafael, California.

Fluid Dynamics - W. T. Ivey, Statistician & Planning Engineer, Southern Natural Gas Company, P. O. Box 2563, Birmingham 2, Alabama.

Note: No. 1957-6 is part of the copyrighted Journal of the Pipelines Division of the American Society of Civil Engineers, Vol. 83, PL 1, March, 1957.

Pipeline Crossings of Railroads & Highways - J. E. Thompson, Supt. of Pipeline Construction, Natural Gas Pipeline Co. of America, 122 South Michigan Ave., Chicago 3, Illinois.

Pipeline Location, Surveying & Mapping - H. M. Hayes, Profect Engineer, Ford, Bacon & Davis, Inc., P. O. Box 1762, Monroe, Louisiana.

Pipeline Design, Specifications & Operating Standards - C. L. Shea, Senior Project Engineer, Products Pipelines, Shell Oil Company, 4459 North Keystone Avenue, Indianapolis 5, Indiana.

Pumping & Compressor Stations - F. E. Culvern, Research Engineer, Panhandle Eastern Pipeline Co., 1221 Baltimore Ave., Kansas City 5, Mo.

Storage of Pipeline Fluids - chairman to be appointed.

Executive Committee:

Chairman E. V. Hunt, Chief Engineer, The Alberta Gas Trunkline Co. Ltd., 320 - 9th Avenue West, Calgary, Alberta, Canada.

Vice Chairman A. E. Poole, President, The Hallen Construction Co., Inc., 4270 Austin Boulevard, Island Park, New York.

S. E. Huey, S. E. Huey & Company, Heninger Building, Monroe, Louisiana.

F. C. Culpepper, Jr., Project Manager, Ford, Bacon & Davis Construction Corporation, P. O. Box 710, Alliance, Ohio.

Secretary J. B. Spangler, Transcontinental Gas Pipe Line Corporation, P. O. Box 296, Houston 1, Texas.

Contact Member from Board of Direction, L. A. Elsener, V. P., Chicago Bridge & Iron Company, 100 Bush Street, San Francisco 4, Calif.

PROCEEDINGS PAPERS

The technical papers published in the past year are identified by number below. Technical-division sponsorship is indicated by an abbreviation at the end of each Paper Number, the symbols referring to: Air Transport (AT), City Planning (CP), Construction (CO), Engineering Mechanics (EM), Highway (HW), Hydraulics (HY), Irrigation and Drainage (IR), Power (PO), Sanitary Engineering (SA), Soil Mechanics and Foundations (SM), Structural (ST), Surveying and Mapping (SU), and Waterways and Harbors (WW) divisions. Papers sponsored by the Board of Direction are identified by the symbols (BD). For titles and order coupons, refer to the appropriate issue of "Civil Engineering." Beginning with Volume 82 (January 1956) papers were published in Journals of the various Technical Divisions. To locate papers in the Journals, the symbols after the paper numbers are followed by a numeral designating the issue of a particular Journal in which the paper appeared. For example, Paper 1113 is identified as 1113 (HY6) which indicates that the paper is contained in the sixth issue of the Journal of the Hydraulics Division during 1956.

VOLUME 82 (1956)

MARCH: 906(WW1), 907(WW1), 908(WW1), 909(WW1), 910(WW1), 911(WW1), 912(WW1), 913 (WW1)[c], 914(ST2), 915(ST2), 916(ST2), 917(ST2), 918(ST2), 919(ST2), 920(ST2), 921(SU1), 922(SU1), 923(SU1), 924(ST2)[c].

APRIL: 925(WW2), 926(WW2), 927(WW2), 928(SA2), 929(SA2), 930(SA2), 931(SA2), 932(SA2)[c], 933(SM2), 934(SM2), 935(WW2), 936(WW2), 937(WW2), 938(WW2), 939(WW2), 940(SM2), 941 (SM2), 942(SM2)[c], 943(EM2), 944(EM2), 945(EM2), 946(EM2)[c], 947(PO2), 948(PO2), 949(PO2), 950(PO2), 951(PO2), 952(PO2)[c], 953(HY2), 954(HY2), 955(HY2)[c], 956(HY2), 957(HY2), 958 (SA2), 959(PO2), 960(PO2).

MAY: 961(IR2), 962(IR2), 963(CP2), 964(CP2), 965(WW3), 966(WW3), 967(WW3), 968(WW3), 969 (WW3), 970(ST3), 971(ST3), 972(ST3)[c], 973(ST3), 974(ST3), 975(WW3), 976(WW3), 977(IR2), 978(AT2), 979(AT2), 980(AT2), 981(IR2), 982(IR2)[c], 983(HW2), 984(HW2), 985(HW2)[c], 986(ST3), 987(AT2), 988(CP2), 989(AT2).

JUNE: 990(PO3), 991(PO3), 992(PO3), 993(PO3), 994(PO3), 995(PO3), 996(PO3), 997(PO3), 998 (SA3), 999(SA3), 1000(SA3), 1001(SA3), 1002(SA3), 1003(SA3)[c], 1004(HY3), 1005(HY3), 1006 (HY3), 1007(HY3), 1008 (HY3), 1009 (HY3), 1010 (HY3)[c], 1011 (PO3)[c], 1012 (SA3), 1013 (SA3), 1014(SA3), 1015(HY3), 1016(SA3), 1017(PO3), 1018(PO3).

JULY: 1019(ST4), 1020(ST4), 1021(ST4), 1022(ST4), 1023 (ST4), 1024(ST4)[c], 1025(SM3), 1026 (SM3), 1027(SM3), 1028(SM3)[c], 1029(EM3), 1030(EM3), 1031(EM3), 1032(EM3), 1033(EM3)[c].

AUGUST: 1034(HY4), 1035(HY4), 1036(HY4), 1037(HY4), 1038(HY4), 1039(HY4), 1040(HY4), 1041(HY4)[c], 1042(PO4), 1043(PO4), 1044(PO4), 1045(PO4), 1046(PO4)[c], 1047(SA4), 1048 (SA4)[c], 1049(SA4), 1050(SA4), 1051(SA4), 1052(HY4), 1053(SA4).

SEPTEMBER: 1054(ST5), 1055(ST5), 1056(ST5), 1057(ST5), 1058(ST5), 1059(WW4), 1060(WW4), 1061(WW4), 1062(WW4), 1063(WW4), 1064(SU2), 1065(SU2), 1066(SU2)[c], 1067(ST5)[c], 1068 (WW4)[c], 1069(WW4).

OCTOBER: 1070(EM4), 1071(EM4), 1072(EM4), 1073(EM4), 1074(HW3), 1075(HW3), 1076(HW3), 1077(HY5), 1078(SA5), 1079(SM4), 1080(SM4), 1081(SM4), 1082(HY5), 1083(SA5), 1084(SA5), 1085(SA5), 1086(PO5), 1087(SA5), 1088(SA5), 1089(SA5), 1090(HW3), 1091(EM4)[c], 1092 (HY5)[c], 1093(HW3)[c], 1094(PO5)[c], 1095(SM4)[c].

NOVEMBER: 1096(ST6), 1097(ST6), 1098(ST6), 1099(ST6), 1100(ST6), 1101(ST6), 1102(IR3), 1103 (IR3), 1104(IR3), 1105(IR3), 1106(ST6), 1107(ST6), 1108(ST6), 1109(AT3), 1110(AT3)[c], 1111(IR3)[c], 1112(ST6)[c].

DECEMBER: 1113(HY6), 1114(HY6), 1115(SA6), 1116(SA6), 1117(SU3), 1118(SU3), 1119(WW5), 1120(WW5), 1121(WW5), 1122(WW5), 1123(WW5), 1124(WW5)[c], 1125(BD1)[c], 1126(SA6), 1127 (SA6), 1128(WW5), 1129(SA6)[c], 1130(PO6)[c], 1131(HY6)[c], 1132(PO6), 1133(PO6), 1134(PO6), 1135(BD1).

VOLUME 83 (1957)

JANUARY: 1136(CP1), 1137(CP1), 1138(EM1), 1139(EM1), 1140(EM1), 1141(EM1), 1142(SM1), 1143(SM1), 1144(SM1), 1145(SM1), 1146(ST1), 1147(ST1), 1148(ST1), 1149(ST1), 1150(ST1), 1151(ST1), 1152(CP1)[c], 1153(HW1), 1154(EM1)[c], 1155(SM1)[c], 1156(ST1)[c], 1157(EM1), 1158 (EM1), 1159(SM1), 1160(SM1), 1161(SM1).

FEBRUARY: 1162(HY1), 1163(HY1), 1164(HY1), 1165(HY1), 1166(HY1), 1167(HY1), 1168(SA1), 1169(SA1), 1170(SA1), 1171(SA1), 1172(SA1), 1173(SA1), 1174(SA1), 1175(SA1), 1176(SA1), 1177(HY1)[c], 1178(SA1), 1179(SA1), 1180(SA1), 1181(SA1), 1182(PO1), 1183(PO1), 1184(PO1), 1185(PO1)[c].

MARCH: 1186(ST), 1187(ST), 1188(ST), 1189(ST), 1190(ST), 1191(ST), 1192(ST)[c], 1193(PL), 1194(PL), 1195(PL),

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c. Discussion of several papers, grouped by Divisions.

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